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More mountings for change gears

Zinc die-castings

Latest developments in beryllium alloys

PRODUCT ENGINEERING

Devoted to the Interests of Engineers and Executives Responsible for Product Design and Development

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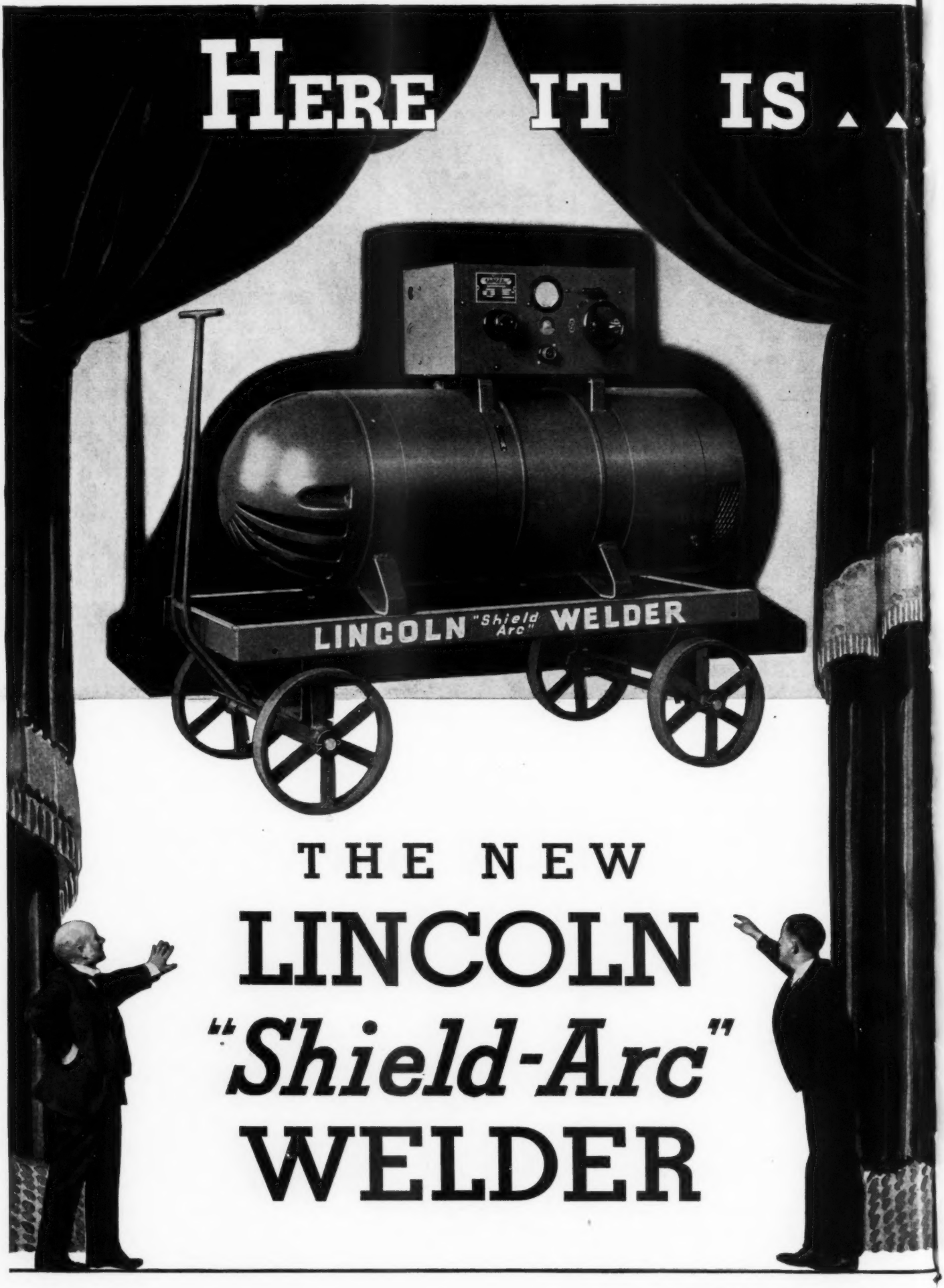
HERE IT IS . . .
LINCOLN "Shield Arc" WELDER
THE NEW
LINCOLN
"Shield-Arc"
WELDER

PRODUCT
ENGINEERING

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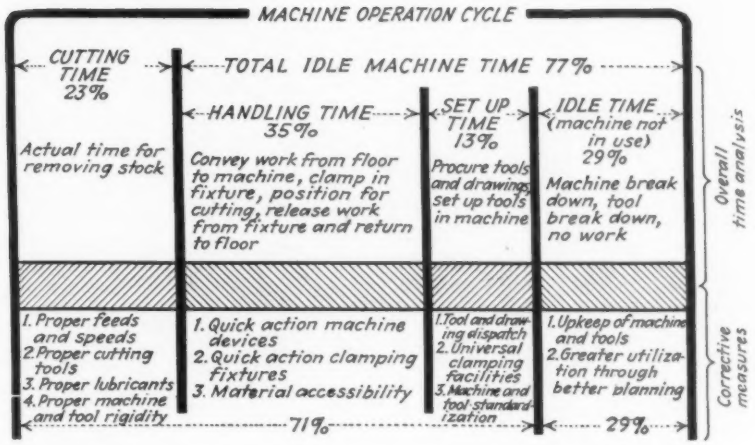
AUGUST, 1932


MACHINE OPERATION CYCLE
CUTTING TIME 23%
TOTAL IDLE MACHINE TIME 77%
HANDLING TIME 35%
SET UP TIME 13%
IDLE TIME (machine not in use) 29%
Actual time for removing stock
Convey work from floor to machine, clamp in fixture, position for cutting, release work from fixture and return to floor
Procure tools and drawings, set up tools in machine
Machine break down, tool break down, no work
Overall time analysis
1. Proper feeds and speeds
2. Proper cutting tools
3. Proper lubricants
4. Proper machine and tool rigidity
1. Quick action machine devices
2. Quick action clamping fixtures
3. Material accessibility
1. Tool and drawing dispatch
2. Universal clamping facilities
3. Machine and tool standardization
1. Upkeep of machine and tools
2. Greater utilization through better planning
Corrective measures
71%
29%

This chart shows actual average percentages for all machines in a large plant. Of the 12 "corrective measures," 5 call for design improvements

Motion Study in the Design of Production Equipment

ALLAN H. MOGENSEN
Assistant Editor
Factory and Industrial Management

An authority on motion study in all its phases, president of the Metropolitan Section of the Society of Industrial Engineers, author of the forthcoming book "Common Sense Applied to Motion and Time Study," Mr. Mogensen is well equipped to write on this subject

▼

It is not very complimentary to the designers of machine tools when one sees the numerous changes that are made to existing equipment in shops where motion study is applied—particularly when one hears suggestions for entire redesign of the equipment. It almost seems as if the men who design this equipment have absolutely no conception of the motions the operator must go through in order to use it.

When machines have to be tended, two separate sets of motions must be provided for: (1) the set which the worker uses when he is tending the machine; (2) the set which the worker uses to prepare tools and material for the machine while it does not require his attention. All machines have to be tended more or less; even automatic machinery has to have attention. It is most important here to have motion study because the earning value of the machine is lost while it is shut down.

Occasionally one sees a machine which can have every and any lever operated without the operator's taking a single step, but comparatively few machines are constructed with this principle in mind.

Machines requiring constant starting and stopping and hand feeding or adjusting should have their various levers so positioned that the laws of motion economy are complied with. Yet in most instances these levers are arranged where they are kinematically or mechanically the simplest. A few of the newer milling machines indicate proper thought along this line, but as yet very little has been done to permit the operator to apply principles of motion economy.

Two devices that are coming into more general use for the elimination of waste motions and idle time—of both men and machines—are the electric time meter and the motion-picture camera. Once we know that waste does exist, and we find out just what and where it is, we have little difficulty in finding means to eliminate it.

The electric time meter records the time that a machine is actually operating. Installed as shown in Fig. 1,

it answers the questions often raised but seldom correctly answered concerning this matter of idle time. So far the use of this instrument has been confined mainly to industrial plants where information as to machine operation with a view to cutting costs was wanted. Although the results of such studies call for improved machines, management is discovering how to get along with present equipment. Special fixtures and equipment are being designed, and in many instances the machine itself is being rebuilt. New machines are not being bought. And they are not being bought because the designers and builders of this equipment know less about their use and practical shop operation than do the people who have to use them.

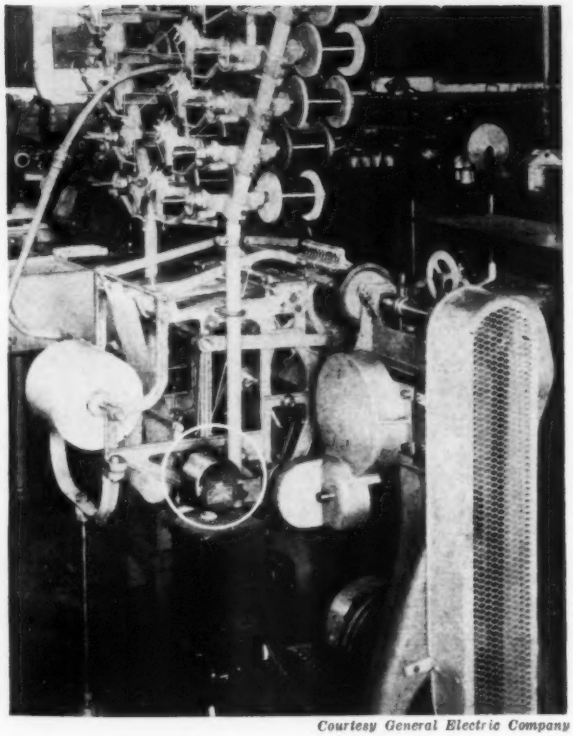
Therefore, not until the designers understand the principles of motion economy and apply them to every tool, fixture and machine, looking at proposed designs from the standpoint of the man who has to use that tool or machine day in and day out, will their products find a market. I realize that this is a strong statement, and a serious indictment, but it is true, as the success of several concerns at the present time bears witness.

Two new devices that furnish even more information than the time meter are now in development, and no doubt as industry realizes the necessity for obtaining the facts as to machine operation, others will appear.

The second device I mentioned is the motion-picture camera. Since the introduction of 16-millimeter film by the Eastman Kodak Company in 1923, amateur motion pictures have steadily gained in popularity. This has made possible continued improvement in machines, lenses, lighting equipment and films which has now resulted in a standard of excellence hardly believed possible a few years back.

The making of motion pictures today does not require professional experience. The 16mm. camera is of the spring-driven type. Two small incandescent lamps are all that are needed. Often no artificial light at all is required. Thus micromotion study becomes a practical tool, rather than a strictly laboratory technique.

Aside from its value in studying the related work of the man and the machine, the camera has tremendous possibilities in the study of machine operation. Using slow motion, the mechanical action of high-speed machinery may be studied, and already several instances of this sort have proved the worth of the camera. A case in point is that of a manufacturer of addressing equipment who was having difficulty with a new high-speed model, but the human eye could not detect the trouble. A slow-motion picture showed one pawl that returned slowly at times—and by correcting this the problem was solved.



Courtesy General Electric Company

Fig. 1—Electric time meter, shown in the white circle, as installed on a coil-winding machine for studying its operation with the view to improving its design

▼

Similar opportunities await the manufacturers of hand tools. Very few of these people have as yet sensed the possibility in redesigning standard tools and equipment so that they can be used with motion economy. One example is the spiral type of screw driver used in many plants throughout the country. Nearly all users have told me that the tools now on the market require a longer stroke than is necessary. It would not be difficult for the manufacturer of this tool to equip it with an eye on one end for mounting and to furnish it with the different pitch demanded.

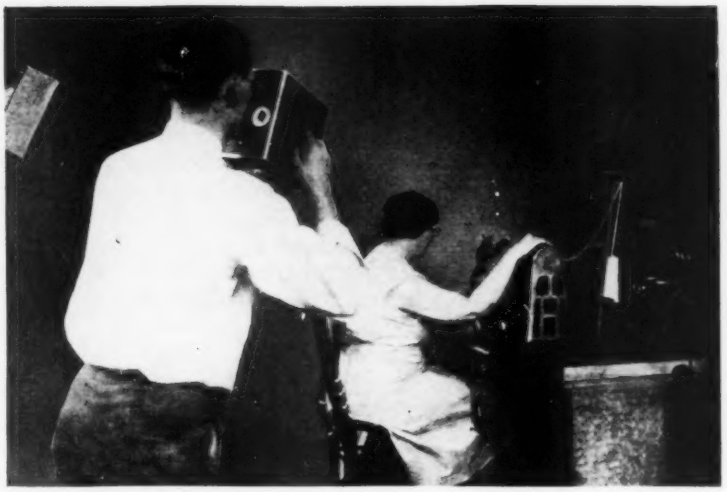
A big opportunity awaits the designer of automatic screw drivers and nut tighteners for small assembly operations, as most of the tools now available are too heavy and too bulky, and I am frequently told by plant men that they have tried these tools and have gone back to the old hand screw driver or open-end wrench.

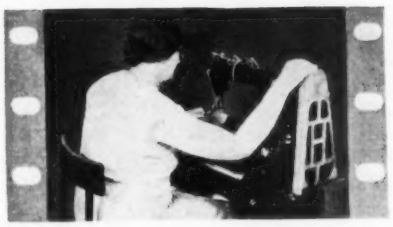
Another example is found in the tools used in tire building. In one plant special tools have been designed which combine two tools in one. This new design, together with pre-positioned holders for the tools, reduces considerable waste in the operation, and yet I have seen this tool in only one plant of the many I have visited.

Another simple example is the split type of screw driver used for picking up and placing small screws. L. P. Persing at the General Electric Company plant in Fort Wayne has shown me many of these small screw drivers. In use they should certainly pay for themselves rapidly. Yet they have to be made in the toolroom and are not generally accepted.

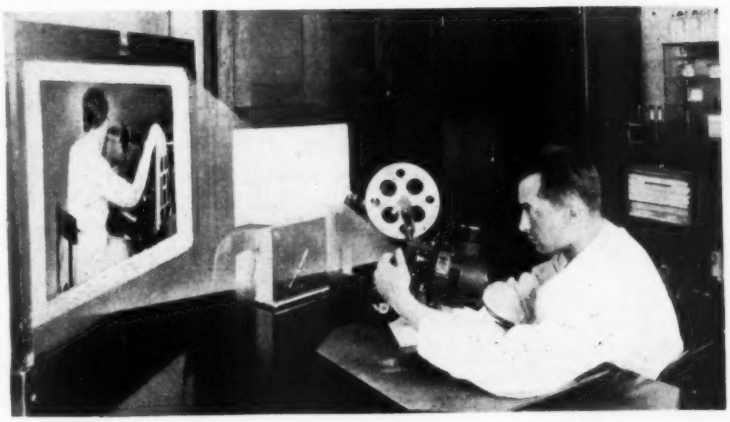
A typical example of changes that make a machine tool more productive is indicated in Fig. 6. These were added by the user, but might well have been part of the original equipment had the designer thought in terms of motions of the operator.

While many simple changes can be made with existing equipment, entire redesign of the machine is often indicated. I saw an interesting application made by a motion-minded organization recently. This was in connection with a large fixed-station vertical turret lathe—considered to be about the last work in production equipment. Surveying the operation of this machine with principles of motion economy in mind, however,





Figs. 2 and 3—Filming a stitching operation for the purpose of a motion study. To the right is shown a sample frame of the picture as taken. The sewing machine is of the usual type; operator must turn handwheel to bring needle down.



showed that considerable time was taken in opening and tightening the chuck. Somewhere around the shop a compressed-air drill was commandeered and mounted directly in front of the operator's station at this machine. This drill was equipped with a square shaft and a sliding collar which could be moved easily into the socket designed to take the ordinary type of wrench. Thus it was possible for the operator to control the drill with the left hand, and to advance and eject this square shaft with the right. A piece of equipment which seemed to be the last word in machine design was improved in operating efficiency by the addition of this simple device.

In a plant where motion study has definitely justified its existence for many years, the sensitive drill presses would hardly be recognized by their makers. The changes came as a natural evolution in the elimination of waste motions. Take an operation where the operator was drilling a small part in a fixture, hand held, in the usual manner. The first step might be to fasten down the drill jig, to permit the left hand to pick up and have ready the next piece, while the right operated the hand lever. Drop delivery might well eliminate the time formerly taken for "put away." The next step might be to move the job to a two-spindle drill, lengthen the levers and equip them with adjustable weights, and have the properly weighted handles alternately feed the drills into the work, the operator loading and unloading the fixtures. In the ultimate development, we can arrange a foot lever to raise both spindles, the operator can load both fixtures simultaneously, and

Figs. 4 and 5—Above is shown the analyst studying the film frame by frame and applying the laws of motion economy. It was found that the operator spends almost half the time turning the handwheel, only one hand being then available to assemble and guide the work. To the right is shown the result of the redesign of the machine on the basis of motion study. A pedal operated mechanism was developed which brings the needle to the down-position when the foot is depressed. Thus both hands are available to guide and assemble the work. Photographs by courtesy of the Hood Rubber Company.

and then permit the weights to drill the two parts while the hands travel to duplicate parts containers now conveniently located. When the spindles are raised by the foot lever, the two finished pieces are ejected into a drop-delivery chute, the two new parts placed in the jig, and the cycle repeated. Over 400 per cent increase in production, and much less effort on the part of the operator.

It can be seen that the first step in improvement was one that could be done and might often be done by a time-study man, but that the later changes actually involved a change in the equipment itself. This sort of thing is not limited to mass production, and can be applied by the small plant as well as the large one.

An instance where entire redesign was necessary was found in a machine that had been built specially for threading the ends of toothpaste tubes. A good deal of money had been spent on this machine, and when completed it was found that production costs were too high. A micromotion study was made of the operation, and when the film was projected it was at once evident that the man was going through contortions like those of a shadow boxer in order to load and unload the tubes. The indexing table consisted of a disk mounted vertically with six pins or mandrels equally spaced around the periphery. The operator bent down to one side, secured the new tube and placed it on a pin at the top of the disk. He then removed the threaded tube from the lower side and placed it in the container for finished parts. Both these boxes were poorly placed, but it was perfectly obvious that no amount of instruction in operation or proper placement of materials could apply good motion practice throughout the operation. The only cure was redesign of the machine.

The equipment was scrapped and a new one built with four pairs of mandrels equally spaced, thus giving a total of eight instead of six. A double threading head was built, threading two tubes at one time. Inclined hoppers at either side made it possible for the operator to be seated comfortably in front of the indexing disk; by the use of simultaneous and opposite motions he was able to load and unload with methods of least waste and fatigue.

Another example occurred in the plant of a concern manufacturing rubber footwear. The operation was stitching the tongue in a canvas basketball or tennis shoe. Wage incentives had been installed and standards set, and the operators seemed to be working actively, but production costs were high. The operation was filmed, as is shown in Fig. 2, and when viewed the trouble was immediately apparent. Too much of the operation time was consumed by one hand reaching up and turning the handwheel of the sewing machine. It was necessary to bring the needle down through the material every time the part being sewed was turned. A pair of scissors used in cutting the threads had to be picked up and put down with the right hand, twice for each shoe. On the film this was so evident that it looked almost ridiculous. A frame from this film is pictured in Fig. 3. The hand can be seen just going to the wheel, while the other holds the work. This delay immediately suggests redesign of the equipment.

Redesign of the sewing machine replaced the hand wheel with a chain and clutch arrangement, whereby pressure on a small foot control automatically brought the needle down through the material. Both hands could then be used to position the part, and a suitable cutting device replaced the large awkward scissors. Thus the operation was mainly one of sewing, using the machine efficiently, rather than a sequence of awkward and fatiguing hand motions that added nothing of value to the part.

Figure 5 is a frame from the "after" film. Here can be seen the guard covering the new device on the machine head. The guard was made necessary because of the habit tendency on the part of all operators who had ever run the standard sewing machine to reach for the old hand wheel. It will be noted on close examination that the operator is using both hands for the operation. Thus she can assemble, position, guide and cut the parts, using the machine to fuller advantage. While this change seemed perfectly obvious, I have not as yet seen any attempt on the part of the builders of this equipment to incorporate it as standard practice.

These are but a few examples of many that could be given emphasizing the necessity for a knowledge of motion study on the part of designers of tools and equipment.

We all know the influence of proper tools upon output, and in the reduction of manufacturing costs. And yet, when we see how little attention has been given to development of tools and equipment in any line from the viewpoint of motion economy, we cannot help but feel that here lies one of the greatest opportunities for development.

Men who can apply these principles will be in demand. And if the present designers and engineers do not meet these demands, others who do not yet know that "it can't be done" will take their place.

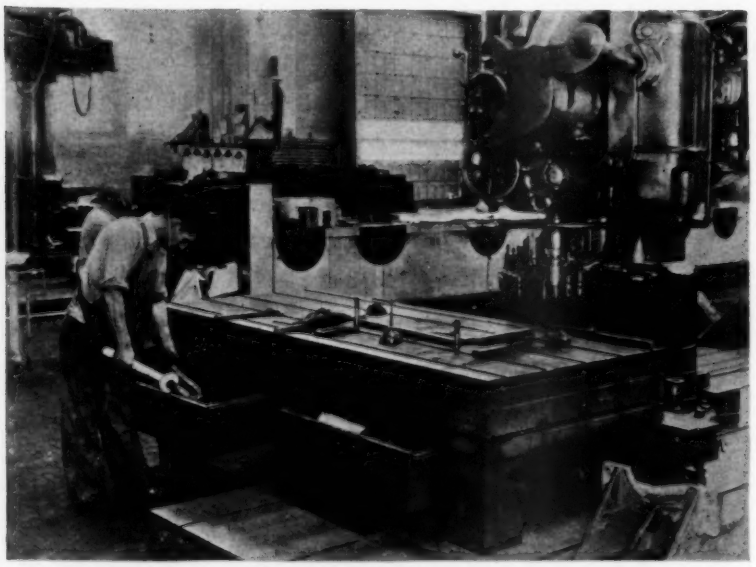


Fig. 6—The designer of the machine failed to do it, the user did it to reduce idle time; drawers under the table for storing clamps and clamping bolts while the column is fitted with a radial holder for drills, reamers and sockets

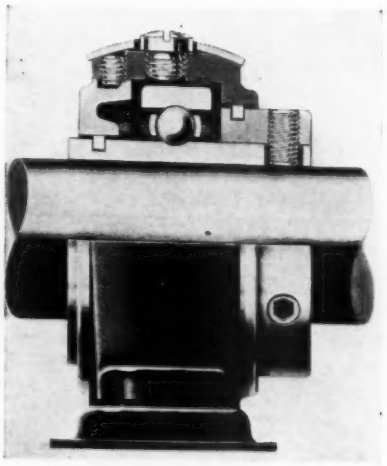
Product Developments

▸ New Applications of Parts and Units

In all the models of a new reflector-type electric street railway car heater, Railway Utility Co. announces the adoption of replaceable Chromalox inclosed electric heating units.

In a motor-driven engine lathe just put on the market by Rockford Machine Tool Co. the designers have incorporated a Westinghouse-Wise drive so that spindle speeds may be changed without stopping the motor. Incidentally the complete job will sell at a price approximately that of a cone-head lathe.

Consolidated Aircraft Co. utilizes over 100 Fafnir ball bearings in the control equipment of its model 21 trainer. Double-seal types, cartridge types, narrow series, torque tube designs find their places in the elevator, rudder and aileron hinges and controls, guide pulleys, clevis mast attachment points, landing gear joints and stabilizer control.



Grease seals of piston ring type protect this new type DH-1 Dodge ball bearing pillow block against lubrication leakage and dirt

Sheet cork, 3-in. thick and sealed with hot asphalt is used for insulating the new Larkin-Warren "cube-maker." This electric refrigerator is intended for ice production alone and has a capacity of 1,980 ice cubes per 24 hours, 252 lb. in all. 432 cubes freeze at a time in 18 trays. Specially designed 35-cube, waxed paper storage cartons increase the storage capacity of the unit.

Jas. P. Marsh Corp. has designed the whole bonnet assembly of this valve as a replacement unit to modernize old radiator inlet valves. Assembly is packless and of quick-opening high-lift construction

Four chrome-cobalt magnets, whose permanency is claimed to be 40 years, constitute the active elements for catching iron particles, in an automobile oil filter announced by Simms Motor Units, Ltd., London, England.

With photoelectric control, Package Machinery Co., has designed a machine for wrapping packages with printed transparent cellulose materials cut automatically from a continuous roll. If the control dot on the wrapper is over the photocell when the material stops for the knife, the printing is in the proper position, and no adjustment is made. But if the dot is ahead of the photocell, so that light is admitted to it from the lamp on the opposite side of the wrapper, the relay goes into instantaneous operation, and the "corrector" operates to reduce the length of the next wrapper.

Heliocentric speed reducer designed as integral part of conveyor pulley by Universal Gear Corp. The pulley drum constitutes the reducer case. In operation the output shaft is locked stationary so that when the input is driven the reducer and pulley revolve at reduced speed

Said to be the first commercial application, an electric motor drives a Bell & Howell motion picture camera designed for airplane and scientific work.

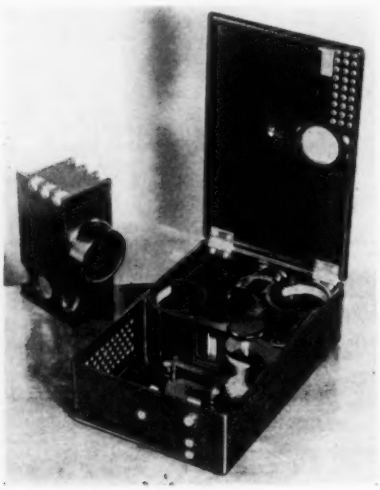
Thompson Products, Inc., utilizes Westinghouse photocells in heat-treating furnaces for automotive valves. An electric heating current passes through the valve itself. When the proper temperature is reached, the valve glows with a certain precise color; the electric eye acts immediately and with precision. Through a suitable relay it shuts off the current and automatically drops the valve into an oil quench.

A new General Electric hysteresis synchronous motor is said to be the first governorless type self-starting synchronous motor ever applied to a phonograph. The armature is composed of a plain cylinder of laminated permanent magnet steel. Since the motor speed is just as constant as the frequency of the power supply, the turntable runs at constant speed.

Electronic tubes make possible a bell-less belfry. In the 49-note carillon, developed by RCA-Victor, tiny hammers strike small tuned metal reeds. A 1,000-watt amplifier takes the musical vibrations from the thin reeds and sends them out as bell tones through a sound-projector in a distant tower. Operation is by hand or by music roll.

▸ New Materials Uses

Cellulose acetate containers for liquids and solids are in production by the Cleveland Container Co. These have water-tight metal bottoms crimped on to two-ply transparent, spiral-wound cellulose acetate sheets, the latter ranging in thickness from 0.002 in. to 0.007 in.



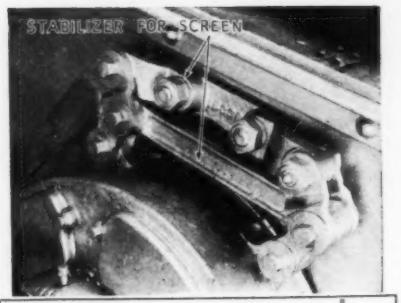
Probably the most intricate piece ever molded out of Durez, this new still-film projector is an assembly of four separate pieces produced by Norton Laboratories, Inc. Only finishing operation is removal of slight fins

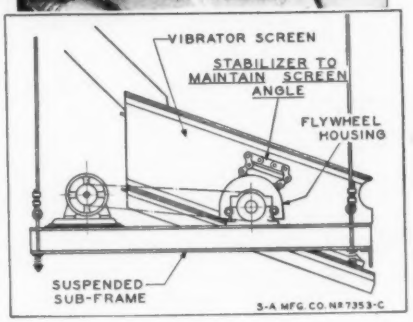
Aluminum or alloys thereof are reported in a dozen recent applications: drawing instruments for draftsmen; 1-lb. and $\frac{3}{4}$-lb. machinists' hammers; heads for squaring shears; aluminum fins on copper evaporation coils for mechanical refrigerators; also cast and extruded sections as refrigerator "ice fins"; milk can for transport from farm to dairy; truck rear axle housing; ram for machine shop shaper to replace cast iron; crumpled foil as insulator for refrigerator truck body; mine cars one-third lighter than steel predecessors and requiring no painting; friction head on hoists permitting higher speed operation; lawn mowers.

Pieces of 6- or 8-lb. sheet lead are being used as vibration dampeners for lubrication pipe lines on pumps and other machines. These are inserted between the pipe and the clamp.

Tungsten carbide is the material adopted for the ends of spindles and anvils of a new line of Brown & Sharpe micrometer calipers.

From Germany comes a coffee grinder with case and handles of Bakelite and a Bakelite spectacle case with an unusual concealed hinge and cover clamp. Other recent applications of this phenol plastic are the base and handle of a Manning-Bowman fruit juice reamer and the colored handle of a Conrad safety razor. The latter comes in a range of colors, the thought being that the razor, like the tooth brush, should serve but one master.



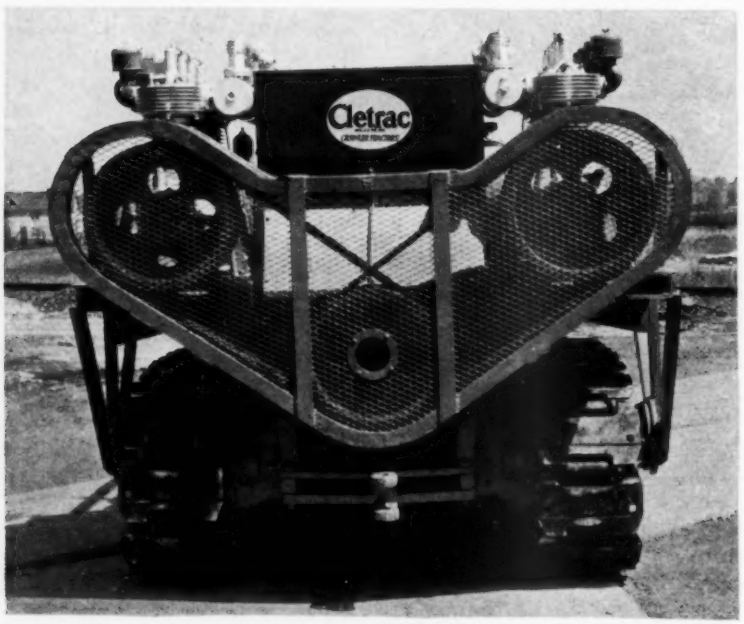


Micarta, decorated with highly polished aluminum inlays, veneers the sides of the new steel water cooler recently introduced by Westinghouse Electric & Mfg. Co.

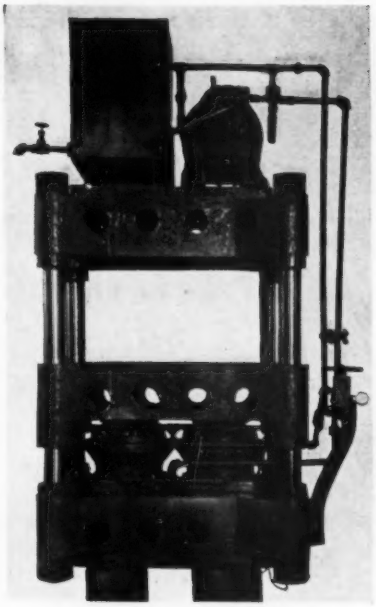
Rubber cushions the frames of certain shop trucks being produced by The Service Caster and Truck Co. The understructure of one model is cast aluminum with frame and platform of rolled aluminum; the handle is aluminum pipe.

▸ Mechanical Stabilizer

In ordinary vibrating screen design the screen body is balanced upon springs which hold it at the proper screening angle and yet are flexible enough to permit its vibration. But



New tandem air compressor mounting does not interfere with drawbar work. Cleveland tractor mounts tandem Davey compressors in such equilibrium that the whole machine is claimed to operate successfully at an angle of 40 deg. in any direction. Compressor drives are independent sets of V-belts



the designers of Stephens-Adamson Mfg. Co. are replacing the springs with a single mechanical unit called a "stabilizer." As shown in the illustrations this is mounted on one side of the screen, and while the screen body can vibrate freely with the eccentric drive shaft, it cannot be rocked or bounced by sudden surge of material over the screen.

The stabilizer consists of two pairs of cast steel arms, each jointed in the middle and held parallel by a heavy cross bar. One end of each arm pivots on the flywheel housing which is solidly mounted on the steel subframe. The other end of each arm is free to move in a shackle on the screen body. The arms are free to swing or bend in any direction, but both must act together. Even in screens having a long throw, the movement of the stabilizer arms is so slight that there is very little wear on any part.

▸ Self-Contained Molding Press

This two-cylinder, moving-up type press, one of several recent installations for plastic molding, designed and built by the Lake Erie Engineering Corp., has the unique feature of being a self-contained motor-driven unit. The low pressure stage of a special two-stage rotary pump closes the molds rapidly; the high pressure comes into play automatically. The press is equipped with pullbacks to insure positive return of the moving platen and to actuate the strippers. With valve control by a simple single-stem four-way valve of special design, this self-contained design eliminates the necessity of an accumulator. Although the press is of large tonnage, pressures can be regulated from 250 lb. to 2,500 lb. to cover a wide range of work.

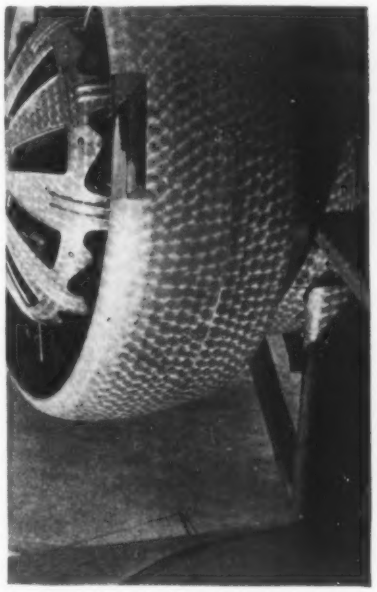
▸ New Furnace Design

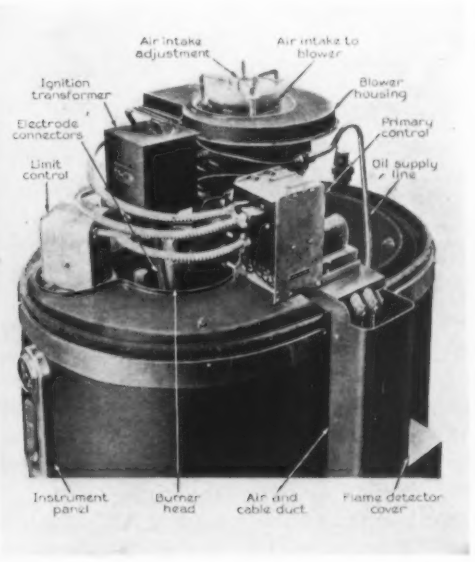
The new oil furnace for home heating recently put on the market by General Electric Co., reflects some interesting features of design. Apparently following out the same general idea as used in the design of its electric refrigerator, the upper section of the furnace, completely heat insulated from the lower sections, contains the control instruments or devices and the oil burner mechanism. All rotating parts are mounted on the extended shaft of the motor, which has no brushes and is probably of the capacitor type, and all lubrication is under pressure by a sealed-in oil supply.

Oil burner and nozzle and electrodes for electric-arc ignition project from upper chamber down into combustion space through sealed openings, being kept cool by a small auxiliary air supply. Main air supply from blower passes through a duct on left side of furnace, and is then directed upwards from bottom of combustion chamber. Flame path is down the centre of the combustion chamber, mushrooming out when it meets the main air supply, and then passes up along the furnace walls. From the top of the combustion chamber the hot gases sweep through a short lateral pass and then down over fabricated boiler tubes. Flue connection to the chimney is at the bottom of the boiler.

▸ Mirrors for Stroboscopes

On tri-motored airplanes, pilots have been forced to synchronize their engines almost entirely by ear, adjusting the throttles until the disappearance of audible "beats" indicates that the vibration periods of all engines are in phase. But as shown in the illustration, Chief Pilot George Kraigher, of the western division of Pan American Airways, has perfected a method of visual synchronization





by means of small mirrors of highly polished metal attached to the inboard leading edges of the ring cowlings on each nacelle motor.

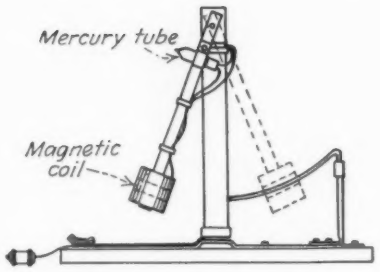
These mirrors are so placed that the pilot can see a reflection of the nose engine and propellor through the rotating outboard propellor. The image of the nose propellor is thus superimposed upon the disk of the outboard propellor, and any difference in r.p.m. shows up as visual beats. The direction of rotation of the beat image in the mirror shows whether the outboard engine is turning up faster or slower than the center engine. Using the center image as a master and adjusting the throttles of the outboard until the image of the propellor in the mirror appears stationary, the pilot may synchronize all three engines.

▸ Humidity Control

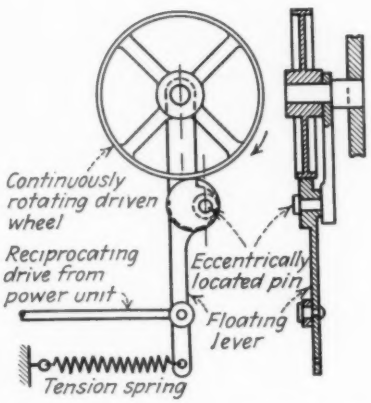
With an airplane type fan and a nickel-silver water disk rotating in dynamic balance at opposite ends of a $\frac{1}{3}$ hp. motor driven shaft, the Southworth Humidifier, manufactured by Southworth Machine Co., will automatically hold relative air humidity in factories and laboratories practically stationary. Excepting for the water valve and control, which is a special counterweighted cord sensitive within two minutes to a humidity change of as little as 2 per cent, the whole device is enclosed in an open-ended metal shell of venturi tube shape. Largest diameter is 14 in. A single unit is designed to serve 12,000 cu.ft., subject to special estimates for unusual conditions. Since the device is constructed out of corrosion-resistant materials, chlorinated water has no adverse effect upon it. Should an accident occur, an automatic control valve closes instantly and insures against water damage.

▸ Solenoid Motor

The "pendulum power unit" recently brought out by the Specialty Products Division of the Mergenthaler Linotype Company, is remarkable for the extreme simplicity of its design. This "motor" is used for driving advertising display devices and for similar purposes requiring only small amounts of power. As shown in the accompanying line drawing, the power unit is merely a solenoid coil fastened to the lower end of a pendulum. The solenoid plunger is a rigidly mounted solid bar curved to permit the solenoid to swing through the arc of the circle. Make and break of the current is obtained by mercury contact in a glass tube, the globula of mercury running from one end of the tube to the other as the pendulum swings. The only bearing is the pendulum suspension.



"Pendulum Power Unit" developed by the Mergenthaler Linotype Company is remarkable for its extreme simplicity and low power consumption.



Patented mechanism used to convert the oscillating motion of the pendulum power unit into unidirectional rotation.

This power unit can be run on either a.c. or d.c. the action being comparable to that of a magnet and its armature. The solenoid coil is wound for a.c., and when the d.c. current is to be used, the flexible cord for connecting to the power circuit is furnished with an adapter plug that provides the required resistance. The unit consumes an extremely small amount of power, the current flowing only during a fraction of the swing.

The resulting motion is of course an oscillating one, but this can be transformed into a practically continuous rotary motion by means of the drive shown in the figure. The swinging of the pendulum causes the floating lever to oscillate. As can readily be understood from the illustrations, the eccentric on the floating driving lever makes contact with the drive wheel, causing a rotation of the drive wheel when the oscillation is in one direction, but sliding across the face of the drive wheel when the oscillation is in the other direction. The purpose of the tension spring is merely to keep the eccentric in contact with the face of the driven wheel on the idle return stroke.

An interesting feature of this pendulum type motor is that the speed of the motor is solely a function of the length of the pendulum. Using the simple design as shown in the illustration, a pendulum length of about 3 ft. will give one vibration per second. As the pendulum is shortened, the speed increases. However, it is not necessary to resort to an extremely long pendulum in order to get a slow period of oscillation. The same can be obtained by using a weighted pendulum, that is, a pendulum with a weight located above the suspension point.

▸ Rapid Cooling Piston Design

With rapid cooling and maximum bearing-area in mind, the designer of the Dual-Flex aluminum alloy piston supports the head by a concentric ring, or inside wall, close to the center. This supporting wall, which takes the thrust load and contains the pin bosses, joins the skirt at the base of the piston. Heat from the head, therefore, must travel the depth of the piston and then return upward to reach the top of the skirt. United Engine & Machine Co., makers of the piston, claim that the distance of heat travel is six times that of a conventional piston, with a consequent increase in radiating area.

Economical Design of Belt Drives

EMORY KEMLER
Instructor in Mechanical Engineering, University of Pittsburgh

Designing a belt drive, as considered practically and in textbooks is usually one of finding the size of belt to transmit the required amount of power. Usually no attention is given to the economical question of the cost of the belt or drive. The following discussion will point out some interesting facts concerning the design both as to cost and capacity.

The horsepower a belt will transmit is given by:

$$\text{Hp.} = \frac{(T_1 - T_2)\, V \times W}{33000} \qquad (1)$$

T_1 = the tight side tension of the belt in lb. per in. of width
T_2 = the slack side tension of the belt in lb. per in. of width
V = the pitch velocity in ft. per min.
W = width of belt in in.

The relation between T_1 and T_2 is given by:

$$\frac{T_1 - wV^2/g}{T_2 - wV^2/g} = e^{u\theta} \qquad (2)$$

w = the weight of belt in lb. per in. width per ft.
e = the base of naperian logarithms = 2.718
u = the coefficient of friction between the belt and pulley
θ = the arc of contact in radians of the belt on the smaller pulley

Equation (2) can be reduced to:

$$T_1 - T_2 = \left(T_1 - wV^2/g\right)\frac{e^{u\theta} - 1}{e^{u\theta}} \qquad (3)$$

If we let:

$$\frac{e^{u\theta} - 1}{e^{u\theta}} = A$$

$$\text{Hp.} = \frac{(T_1 - wV^2/g)\, A\, V\, W}{33000} \qquad (4)$$

The constant A varies with u and θ. If then the horsepower, tight side tension T_1 pitch velocity, and coefficient of friction are fixed, W depends only upon θ, which is given in degrees by:

Shortening the center distance between pulleys does not necessarily decrease the cost of a belt drive, while reducing the diameters of the pulleys theoretically does not affect the cost of the belt. Relative cost of short-center drives is also analyzed

▼

$$\theta = 180 - 2 \sin^{-1}\left(\frac{D - d}{2c}\right) \qquad (5)$$

D = the large pulley diameter
d = the small pulley diameter
c = the distance between pulley centers

Once the speed ratio, and pulley diameters are fixed, θ will depend only upon c, the center distance. In order to make the belt width a minimum, we have only to make θ a maximum, which means making c large. The belt length L increases with c, being given by:

$$L = 2c + \pi/2\ (D+d) + (D - d)^2/4c \qquad (6)$$

Making L larger will increase the cost of the belt. The problem of determining the economical belt drive becomes one of finding the value of c which will make the cost of the belt a minimum.

No simple general equation can be derived for the economical design of belt drives since the formulas for both θ and L are difficult to work with. However the step by step method is not difficult.

A 10-hp. motor operating at 1200 r.p.m. is to be used to drive a shaft at 200 r.p.m., the motor pulley is to be 10 in. in diameter and a leather belt is to be used for which $(T_1 - wV^2/g) = 60$ lb. per in. of width for a single ply belt, $u = 0.30$, and which costs 15 cents per in. of width per ply per foot length. The procedure is to assume various values of c and calculate θ, A, L, and W, from which the cost can be determined.

In order to shorten the work of calculating W we can rearrange the equation for Hp. and solve for W in terms of A, as follows:

$$\text{Hp.} = \frac{(T_1 - wV^2/g)\, A\, V\, W}{33000}$$

Since $V = 3140$ ft./min.

$$W = \frac{\text{Hp.} \times 33000}{(T_1 - wV^2/g)\, V\, A}$$

$$= \frac{10 \times 33000}{60 \times 3140 \times A} = \frac{1.75}{A}$$

Table I

C	θ	A	W (theoretical) width)	W′ (actual) width)	L	Belt Cost (theoretical)	Belt Cost (actual)
Inches					Feet	Dollars	
300	171	0.59	2.97	3	59.2	26.25	26.64
240	168	0.585	2.99	3	49.3	22.16	22.19
200	165.6	0.58	3.02	3	42.7	19.33	19.22
150	161	0.57	3.07	3¼	34.5	15.87	16.82
100	151.5	0.55	3.18	3¼	26.3	12.53	12.83
80	144	0.53	3.30	3½	23.2	11.14	12.18
60	132	0.50	3.50	3½	20.0	10.50	10.50
40	112	0.44	3.98	4	17.1	10.20	11.26
35	105	0.42	4.16	4½	16.5	10.30	11.15

The value of A can be obtained from Fig. 1 which gives values of A for θ between 90 and 180 deg. and u between 0.15 and 0.60.

$$\text{Belt cost} = W \times L \times 0.15$$

The value of W is that given by the formulas and W' is the commercial size that must be used. The belt costs are calculated for both the theoretical and actual widths. A plot of the belt costs against distance between centers is given in Fig. 2. It shows that for the particular drive the economical center distance is 40 in. This center distance gives a cost of belt of $10.20 as compared with a cost of $22.19 for 240-in. center distance, which is the usual value recommended. As compared with an angle of contact of 165 deg., as is often recommended, it gives a cost of $10.20 as compared with $19.22. A plot of the cost, based on the commercial width necessary shows that a center distance between 40 and 60 in. gives a minimum cost of about $10.50. It is also seen that the cost for a center distance between 80 and 100 or 110 in. gives a practically constant cost. The cost in this instance is about half that for 200 or 240 in. centers.

In designs, practical considerations must be dealt with as well as economical considerations, so from the combined viewpoint it seems that either the drive using four times the larger pulley diameters for the center distance or a minimum of 165 deg. for the arc of contact does not give the most economical drive. While they give a drive that may last a little longer, the added cost as compared with a drive having a center distance of say 100 in. is not justified.

Another factor to be considered is the proper pulley diameters to use. For a given angle of contact for any drive it is seen that Equation (1) will reduce to WV = constant if Hp., angle of contact and T_1 are fixed. Since $V = \pi dN/12$ where N = r.p.m., WV = constant then reduces to Wd = constant.

From Equation (5) it is seen that if θ = constant, then

$$\frac{D - d}{2c} = \text{constant.}$$

If n = speed ratio:

$$\frac{d(n-1)}{2c} = \text{constant.}$$

$$d/c = \text{constant}$$

Then as d is reduced, W must be increased in the same proportion, but c must be decreased in the same proportion as d. Equation (6) can be written in the form:

$$L = 2c. + \pi/2\, d\,(n+1) + \frac{d(n-1)}{4c}$$

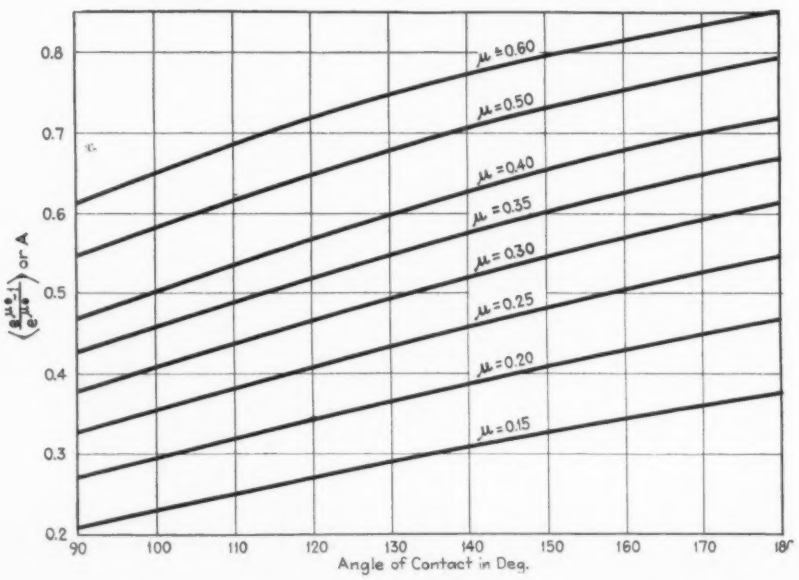
It is seen that if d, for instance, is reduced by half, c will also be reduced by half and therefore L will be cut in half. At the same time, W will be doubled. Since the cost of the belt depends on the product of W and L it is seen that the belt cost remains constant, regardless of this pulley size used. There is, however, usually a saving on the cost of the pulleys. If for instance, a comparison is made between the drive calculated in the example and one using a 5-in. motor pulley and a 30-in. cast iron pulley, it will be noted that the pulley face must be twice that for a 10-in. motor pulley and 60-in. cast iron pulley, since the belt width is doubled. To illustrate the effect of pulley size on costs, the following list prices have been taken from a manufacturer's catalogue. While they do not give actual costs they give relative values and show the effect of pulley sizes.

10 in. dia. × 4 in. motor pulley...	$11.10
60 in. dia. × 4 in. cast-iron pulley	38.20
Total cost.....	$49.30
5 in. dia. × 8 in. motor pulley...	8.50
30 in. dia. × 8 in. cast-iron pulley	20.05
Total cost.....	$28.55

The possibility of using a double thickness belt also enters when the belt width becomes large. However, with a double belt, the stresses induced when the belt is bent over the pulley are increased. The selection of the proper pulley sizes depends upon the prices of the various sizes as well as the belt thickness and it is difficult to draw definite conclusions except in cases where specific data are available.

For open belts, it can be said that standard practice gives a drive that is more expensive than is necessary. It is cheaper to decrease the arc of contact and use a wider belt in order to reduce the belt length. The result is that reducing the angle of contact 15 or 20 deg. below the recommended 165 deg. does not theoretically require a relatively large increase in belt width but does result in a big reduction in belt length. This reduces the cost of the belt a considerable amount. It will be further noted that; (1) a change of horse power, belt tension or belt velocity will not change the economical center distance; (2) the pulley diameter selected does not affect the cost of the belt for any given angle of contact. Any decrease in diameter gives a decreased length and at the same time a proportional increase in belt width, so the cost will remain constant. The cost of the pulleys, however, decreases with a decrease in size and is limited by the

Fig. 1—The value of A, for various coefficients of friction and arcs of contact, can be read directly on this chart



allowable stress resulting from the bending of the belt over the small pulley.

It must be recognized that this analysis of the economical design of belt drives applies only to open belts without idler-pulley tightener. The commercial "short-center drive" presents an entirely different problem. For such designs, the angle θ is determined by the size and arrangement of the pulleys and is practically independent of the center distance. One factor that must not be overlooked in the design of a short-center drive is the stress resulting from the belt bending over the idler if it is the small-diameter pulley. Although the idler pulley is always placed on the slack side of the belt, an excessively small diameter might possibly cause an excessive stress resulting in a relatively short belt life. (See article by same author, "Relation of Belt Stress to Pulley Diameter," P. E. July, 1930, p. 311.)

In order to carry the analysis a step further, the angle of contact for a given "short center" drive can readily be determined from the layout. The chart given in Fig. 1 can then be used to determine the factor A to be used in Equation (4) for solving for W, the width of belt required. Increasing the width to the nearest standard commercial size, and determining the length of belt as called for by the design, the cost can be computed as outlined above.

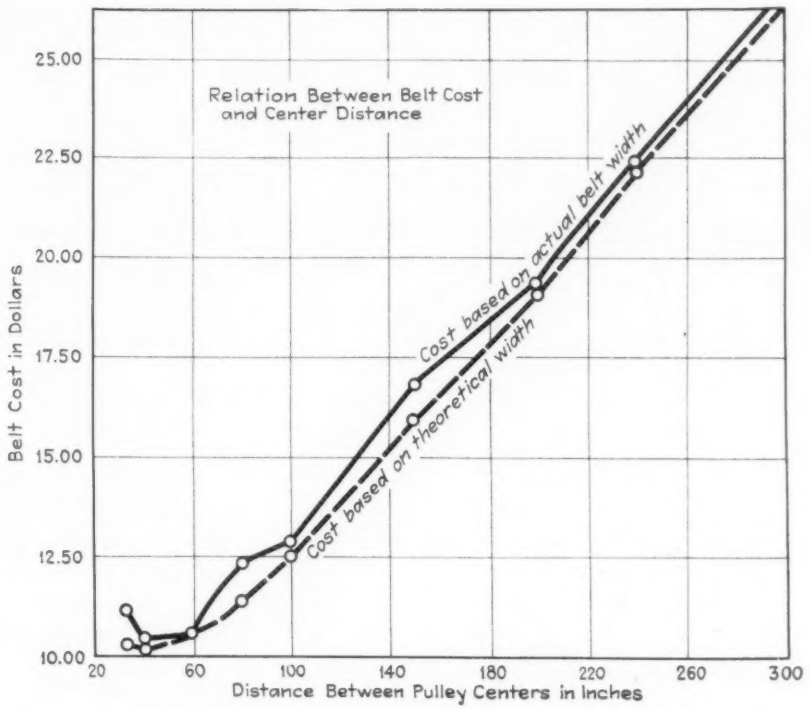


Fig. 2—Relation between belt cost and center distance between pulleys is shown by these curves

Assuming the same general layout as given for the above most economical open-belt drive, namely, 40 in. center distance, 10 in. and 60 in. pulleys, the additional belt length required to accommodate an idler pulley will be about 0.5 ft., depending on the size and arrangement of the idler. This makes the total belt length approximately 17.9 ft. However, with the idler pulley located so as to give 180 deg. of contact between the belt and the motor pulley, and using the same coefficient of friction, namely $u = 0.30$, the value of A will be 0.61. Applying this to the equation for W, we get 3 in. for the nearest commercial width of belt required.

The cost of a 3-in. single ply belt, as assumed in the first example, is 45 cents per foot. Cost of the belt for the short center drive with idler pulley would then be $8.06 as compared to $10.20 for the most economical open belt drive. The difference in belt cost would then be $2.14 in favor of the short-center drive. It must also be considered that this is a recurring saving.

Offsetting the lower belt cost is the expense of the idler pulley and its mounting. This will invariably exceed the saving in the initial cost of the belting, but on the other hand, belt replacement being a recurring expense, the saving over a period of years will be considerably greater. Also, in a plain flat-belt open drive it is usually necessary to provide some form of sliding motor base to take up the stretch. The cost of this should also be applied as a saving to offset the cost of idler and mounting.

Specifying Plating

On account of the throwing power factor, it is not practical to specify that parts containing deep recesses are to be plated, unless it is first investigated to determine whether or not the throwing power of the solution involved will be sufficient to reach the innermost portions. This is particularly true of chromium plating, and is true to varying degrees for all plating solutions.

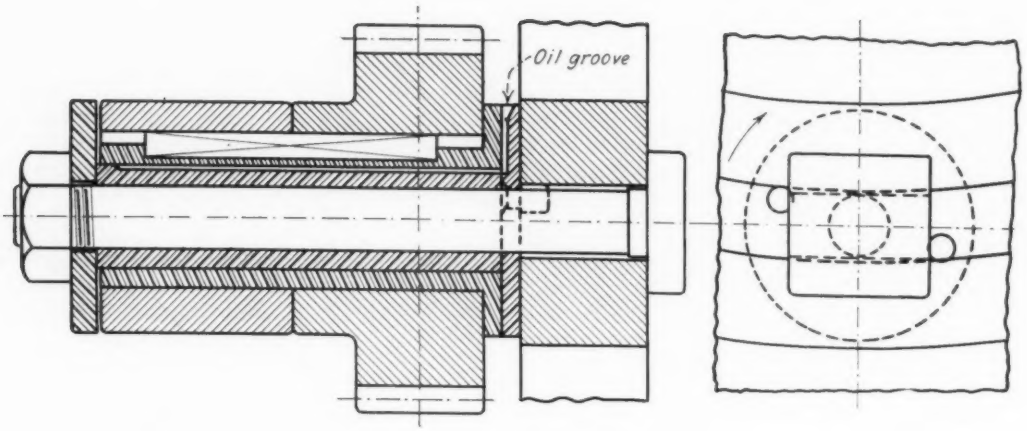
To circumvent this difficulty in the case of stamped and drawn parts, it is sometimes possible to plate the parts prior to the drawing operation. If done properly, there will be no damage to the plated surface as a result of the forming operations.

In discussing this problem J. S. Hoffman of the Udylite Process Company describes how the flat blanked pieces of a condenser frame were cadmium plated in the flat and then formed.

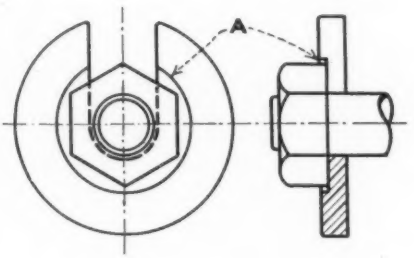
From the standpoint of economical production this method has a great advantage in that the cleaning operations can be performed with much greater efficiency and thoroughness. In view of the fact that a large percentage of faulty plating is caused by improper cleaning, this advantage is obvious.

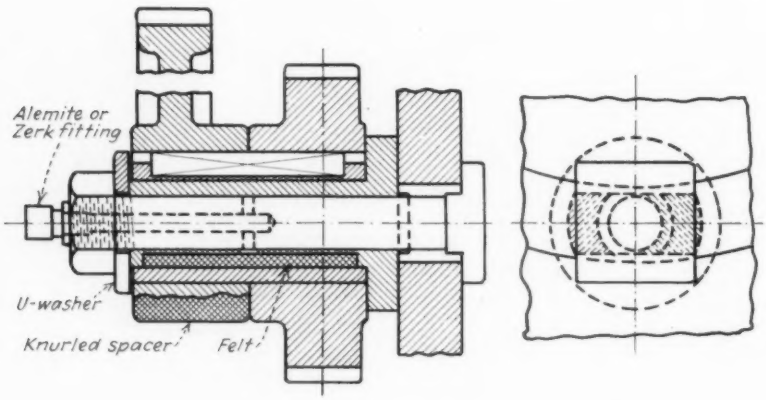
With reference to the metal deposited, plating in the flat gives considerably greater uniformity. Tests have demonstrated that in the case of cadmium plating, the parts are able to withstand severe bending and forming without showing any signs of the coating breaking.

Change-Gear

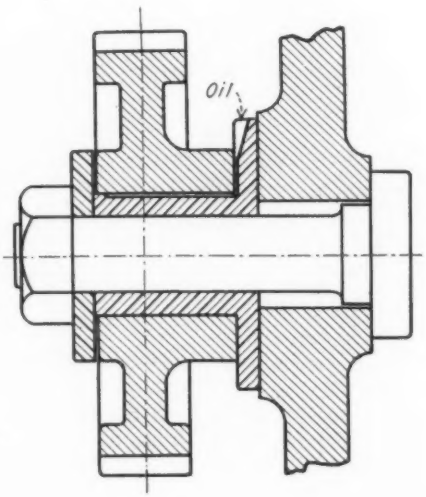


This is an older type of mounting. The innermost sleeve, which must be held stationary, is prevented from rotating by means of the two pins indicated in the end view. This sleeve is made 1/64 in. longer than the gear sleeve in order to obtain the proper running freedom. Lubrication is obtained through the oil hole and groove. The use of a solid washer under the nut necessitates unscrewing the nut completely. The horse shoe type washer shown to the right is better, as it requires the nut to be unscrewed only about 1/16 in. in order to permit the washer to be taken out. The recess *A* under the nut prevents the washer from falling out when it is loosened in order to permit the adjustment of the gears.





In this improved design, the inner sleeve is prevented from rotating by the milled tang or tongue. Also, better provisions are made for lubrication, an Alemite or Zerk type fitting being applied to the end of the stud shaft which is drilled as shown. The lubricant is fed to a felt pad at the under side of the stud and extending not quite to the ends of the gear sleeve. Soft steel on soft steel is sometimes used for the bearing surfaces, but soft steel on hardened steel is a better combination. A further improvement is soft steel and bronze. Studs and sleeves should be designed with generous proportions in order to assure rigidity. Using large diameter studs and sleeves also reduces the unit bearing pressure, thereby improving lubricating conditions and reducing the rate of wear.

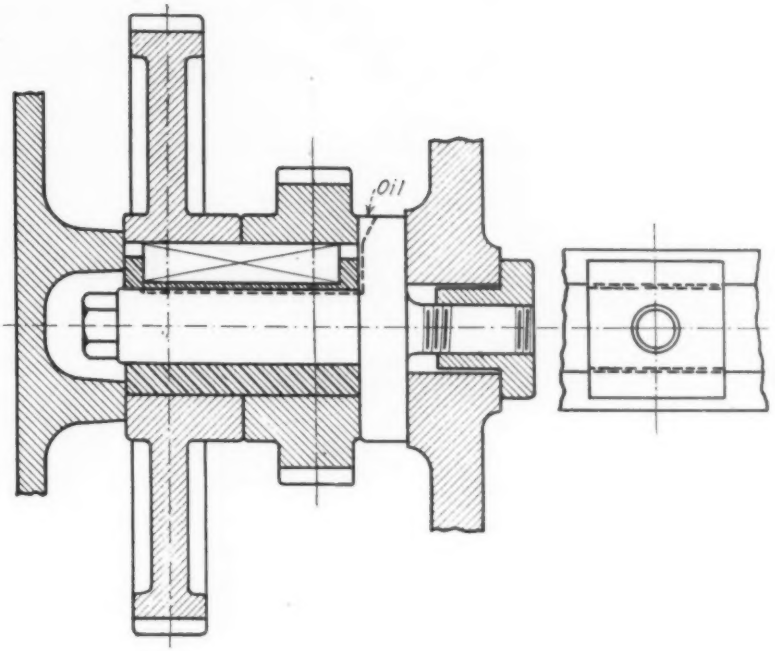


Construction of this reversing idler mounting, though extremely simple, is lacking in qualities required of a good design. The inner sleeve is held from rotating merely by the frictional grip. As the oil hole must be kept in the up position to prevent the lubricant from running out, positive means should always be provided.

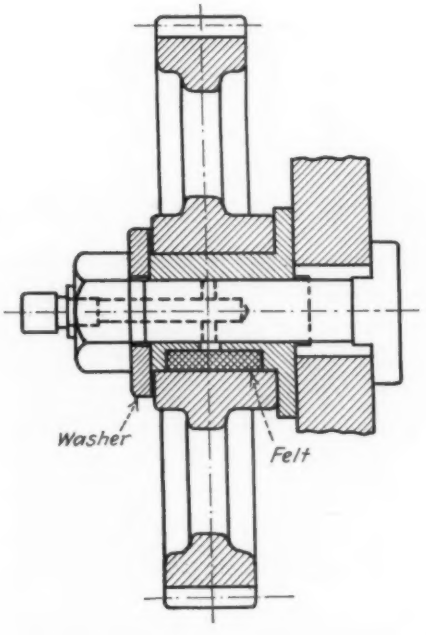
Mountings

ADAM FREDERICKS

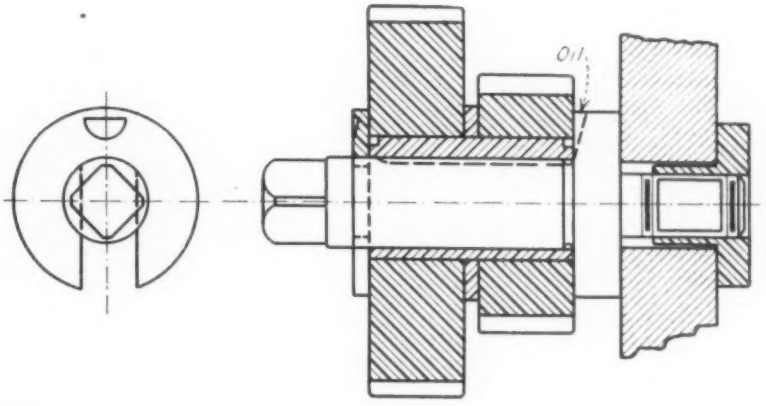
Designs shown here illustrate some tested and successful constructions for mounting gears that are used for speed or feed changes. Additional designs of mountings and gear retaining means will be presented in the next issue of Product Engineering



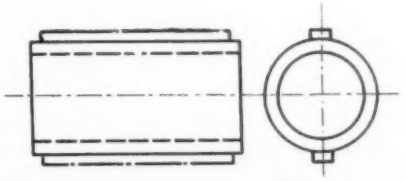
When small diameter gears are to be mounted, there might not be sufficient room for an inner sleeve. The stud can then be screwed into a tee nut and the gear sleeve rotate directly on it. The gears are held in by a cover.



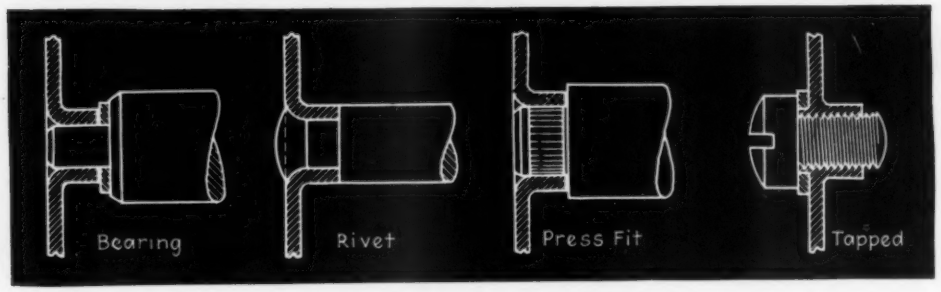
In the design of this reversing idler, a milled tongue prevents the inner sleeve from rotating, the stud shaft is drilled for feeding lubricant to the felt pad, and a lubricant fitting is provided at the end of the shaft. This improved construction is comparable to the second design shown on the opposite page.



For rigidity, the key should be large and have a deep seat. If space is insufficient on account of small bore of gears, the gears can be mounted on a gear sleeve with integral keys. The U-washer has a finger nail slot.



If deemed necessary, two keys can be used when the gear sleeve and key are made integral. The approaching corners and edges of all keys should be rounded to facilitate assembly.



Typical designs for embossings

Embossed Hubs in Stampings

J. K. OLSEN
Chief Draftsman, Stewart-Warner Corporation

In this third article, the author explains how depth of draw, thickness of stock and size of hole all bear a relation to one another. He also considers design factors that affect costs

One of the most common features incorporated in stamped parts is an embossed hub to serve one of four functions: (1) as a bearing; (2) a press fit; (3) for riveting; (4) for a threaded hole. Possible depth of draw with a given thickness of metal at a specified point is the chief problem in designing embossings. It is generally desirable to obtain the maximum length of hub; but for production and die design reasons, the shortest hub is almost always the easiest to make. A number of factors are involved in the selection of the right length: size of hole, thickness of stock, location of embossing in relation to other holes, draws or bends and the edge of the stock. To a certain extent, the sequence of operations must also be considered.

For simple hubs, a practical rule is to make the embossing with a length A equal to half the inside diameter D, as indicated in the accompanying figure. In producing such a hub, ordinarily three steps are involved: (1) punching the hole; (2) forming; (3) sizing. All are performed in a single stroke of the press by a combination punch. There are many parts, however, where a relatively longer length of hub is to be tapped. It is possible to make the length as much as three-quarters the diameter of the hole if there is a sufficient thickness of stock to allow the hub to be drawn out. Much depends upon the absolute dimensions of the embossing. If the hole is extremely small or large, the usual rules do not apply.

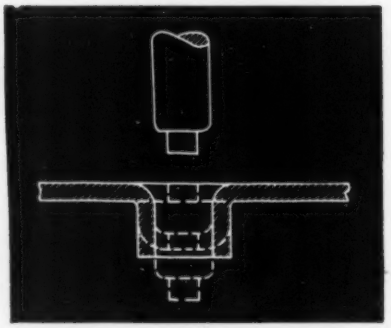
When hole diameter D is less than $\frac{1}{8}$ in., length of hub A can equal $\frac{1}{2}D + \frac{1}{64}$ in.
When D is from $\frac{1}{8}$ to $\frac{5}{16}$ in., then A can equal $\frac{1}{2}D + \frac{1}{32}$ in.
When D is from $\frac{5}{16}$ to $\frac{5}{16}$ in., then A can equal $\frac{1}{2}D + \frac{1}{16}$ in.
When D is more than $\frac{5}{16}$ in., the hub is generally of sufficient length for the purpose intended with $\frac{1}{16}$-in. allowance or with some other proportional allowance.

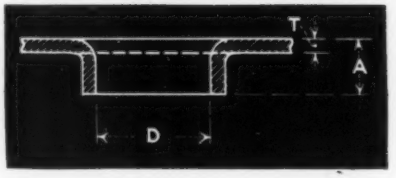
With some experience, the proper length of the hub can be determined by a full-scale layout. But before going too far with a drawn hub design, other proportions should be examined. Often the hole is too small to obtain the length of hub specified without resorting to additional drawing operations, which increase costs. A group of drawn hubs that cannot be made in a one-draw operation is shown in the accompanying figure.

Where a part requires a long drawn hub with a small hole diameter, four or more operations are necessary. In the first draw, a cup is formed slightly larger than the size of the hub; in the second, this cup is drawn back and into the desired shape, making use of the material from the first cup. Two operations may be required to complete this draw, depending upon proportions. In the last two steps the bottom of the cup is perforated and the hub drawn to final size. The problem is largely one of die design, but is mentioned to indicate the cost involved.

Another method for obtaining hubs longer than half the diameter of the

Punching, forming and sizing are all performed in a single stroke of the press by a combination punch





For simple hubs, the practical rule is to make $A = D/2$. As explained in the text, A can be made slightly larger than as given by this proportion

hole includes stretching or squeezing the outside diameter of the hub. In this manner the stock forces its way into the hub, making it longer. When this method is used, the thickness of the stock in the hub decreases in proportion to the reduction in diameter, and there are limitations to the extent to which this can be carried, depending upon the original hardness of the material.

Sometimes the design calls for a hub thicker than usual. An extra operation accomplishes this by setting back a hub drawn to normal proportions. In shortening, the hub thickness and outside diameter are increased. There is a limit to how much the metal in the hub can be successfully set back. If the hub is drawn out too far and the stock thickness decreases too much in the first operation, there is a chance that the stock will fold or buckle in the set-back, instead of a molecular flow or outward swelling taking place. This set-back operation is used to obtain greater surface for an end thrust bearing, or where it is required to use a coarse thread the depth of which makes necessary greater wall thickness.

When the drawn hub is for a threaded hole, additional factors enter. The hole in the hub must be sufficiently smaller than the outside diameter of the screw to provide a suitable overlap of thread in the hub. In practice, this overlap is taken anywhere from 50 to 80 per cent of the depth of thread, usually 75 per cent. Much depends upon the class of thread and its importance in the design. Wall thickness outside the thread should be at least equal to the depth of thread.

For example, let us consider the proportions of a hub to hold a No. 10-32 (0.190 in. O.D.) screw. A 32-pitch screw theoretically has a depth of thread of 0.020 in., and 75 per cent of this would be 0.015 in. Subtracting 0.015 in. from each side would be 0.190 — 0.030, or 0.160 in., the desired size of hole. For practical reasons, the nearest drill rod size, which is 0.161 in., would be chosen. The wall thickness outside the thread would be 0.015 in., the same as the actual depth of thread, and this would give 0.220 as the outside diameter of the hub. Total stock thickness would then be 0.015 + 0.015 = 0.030 in. If the part is made from steel, the next gage size would be 0.032 in. (No. 21 Birmingham), and if made from brass, 0.0319 in. (No. 20 B. & S. gage).

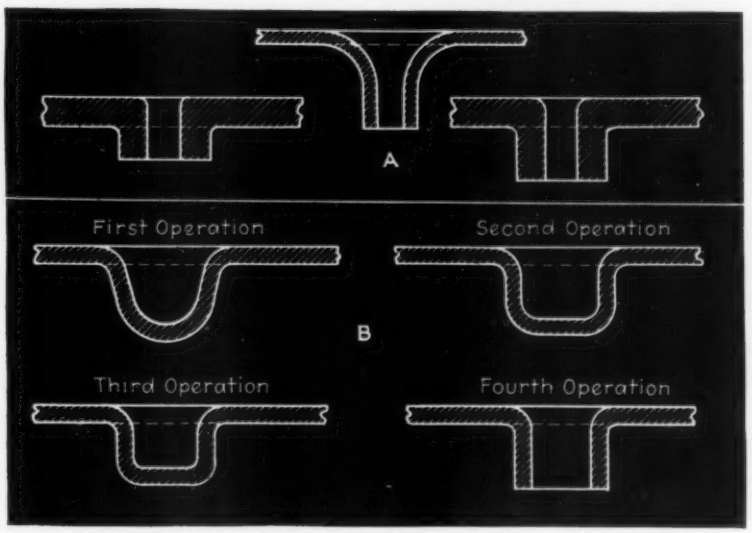
A pipe tap requires a heavier stock thickness to take care of the taper of the pipe thread. To avoid excessive thickness, the hole in the hub can be made to correspond to the pitch diameter at the end of the pipe, to the nearest punch size or fraction. The height of hub is ordinarily taken as the length of the Briggs ring from the end of the pipe, or as the length the gage will go on flush by hand. This height of hub should be measured from the top of the hub to the intersection of the radius where the metal bends up from the base part.

In order to find the stock thickness, account is taken of two pitch diameters—that at the end of the pipe and that at the point where gaging is done, known as the height of the American Briggs ring. Subtracting these diameters and dividing the difference by two gives the amount to be added to the half depth of thread to get the stock thickness used for body material for tapping only. This figure is doubled to get the total wall thickness and should be converted to the nearest gage size. The outside diameter of the hub is equal to the pitch diameter at the end of the pipe plus twice the wall thickness. Actually, the only figures that need be considered are the inside diameter and the thickness of stock. No account is taken of the decrease in stock that takes place in drawing. As this amounts to only a few thousandths of an inch, it will in no way materially affect the strength of the design.

In connection with threaded holes, it is well established by test that the helical ridge of the thread contributes

Dimensions of Drawn Hubs for Pipe Taps

Nominal Pipe Size, In.	Size of Hole, In.	Height of Hub, In.	Stock Thickness Nearest Gage Size		Outside Diameter of Hub, In.	
			Steel	Brass	Steel	Brass
$\frac{1}{8}$	0.368	0.180	0.042	0.045	0.452	0.458
$\frac{1}{4}$	0.469	0.200	0.058	0.057	0.584	0.582
$\frac{3}{8}$	0.625	0.240	0.065	0.064	0.755	0.753
$\frac{1}{2}$	0.750	0.320	0.083	0.0808	0.916	0.911
$\frac{3}{4}$	0.969	0.339	0.083	0.0808	1.134	1.130
1	1.219	0.400	0.095	0.1018	1.408	1.422
$1\frac{1}{4}$	1.563	0.420	0.109	0.1018	1.780	1.766
$1\frac{1}{2}$	1.797	0.420	0.109	0.1018	2.014	2.000
2	2.266	0.436	0.109	0.1018	2.483	2.469



(A) These hubs are out of proportion for a one-draw operation, as there is insufficient metal made available by the hole to provide the volume of metal required for the flange. (B) Embossings that are long relative to the inside diameter necessitate a series of draws, entailing additional expense

to the strength of the embossing an amount equivalent to approximately 10 per cent of the depth of thread.

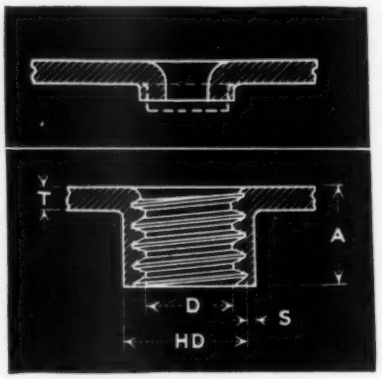
In tapped hubs for mounting screws, the number of threads is an important factor. When the function of the screw is merely to hold the parts together, and no pressure is exerted against or between parts, three full threads may be taken as a safe minimum. In most designs, however, it is desirable to obtain at least four full threads, and the only solution is an extra long hub produced by one of the methods already described or some equivalent method that will permit lengthening the hub.

Other methods are to use heavier stock and retain the same outside diameter of the hub or to increase the size of the hole in the hub and slightly decrease the outside diameter at the expense of the thread depth. Drawn hubs of this kind are surprisingly strong and adaptable for mounting purposes.

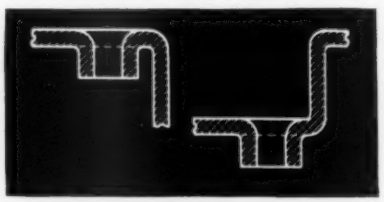
As the metal in a draw must be free to flow, it is objectionable to have the drawn hubs too close to another draw or bend. This may limit the length of hub which otherwise could easily be obtained. Furthermore, it may lead into difficulties in the design of the dies and the production of the part. In the accompanying figure are shown two examples of embossings that are too close to a bend.

Another practical use for embossed metal is for holes in cases and covers of electrical products where cords or wires are passed through. A hub is drawn in the usual way and then formed over to give rounded edges. Where there are only a few of these holes, it may not be economical to use this kind of embossing. Eyelets may be inserted, or for large size holes the sharp corners can be broken by spinning.

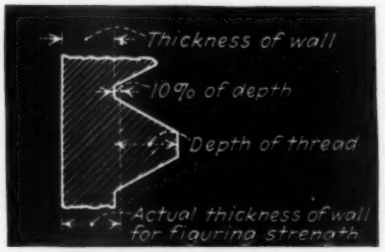
It should be kept in mind that in all drawn parts such as hubs, the drawing operation will decrease slightly the thickness of the stock. As yet there is no way to calculate the amount the metal will stretch. The only practical thing to do is to make a small allowance for this—in most hubs it only amounts to a few thousandths on a side. Where the outside diameter of a hub must be held to close limits, sizing can be done by an added ironing operation.



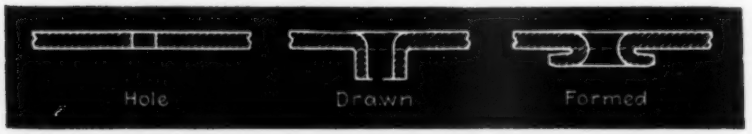
(Above) Extra thickness of the hub for a thrust plate calls for a setback after the first draw. (Below) For a threaded embossing, the stock thickness *T* is governed largely by the depth of thread; while the diameter *D* is governed by the root diameter of the screw



Embossings should not be too close to bends or other drawn sections that would interfere



The thread helix contributes a small amount to the strength of the tapped hub



A type of embossing that provides a hole with rounded edge as is put in sheet metal boxes for electrical instruments for preventing cutting of the leading-in wires

Color versus Form

In his talk before the annual meeting of the Porcelain Enamel Institute, R. Guy Cowan said:

"In the decorative arts there are two channels of expression and consequently two methods of appeal. The first and most important is through color, the second through line and form. Unfortunately all people do not have the capacity to react, they are not all susceptible to these methods of appeal. Color is of greater importance than line, because a larger proportion are sensitive to color. If we could only use music as a method of attraction we would have a larger group capable of interest.

"Briefly I would summarize the capacity to emotional reaction as follows:

To music .....90 per cent
To color.....75 per cent
To line or form...25 per cent

"The percentage of reaction, as between line and color is based on my twenty years experience of making and marketing art products. I believe it is reasonably accurate. If so, then all of us should give about three times as much consideration to color selection as to line and form.

"I find, among the makers of enamel products that too little consideration is given to trend in color, too little consideration to what is offered the public in supplementary equipment and furniture. Taking the kitchen as the principal example, there are several features, anyone of which might set the style and color scheme of the room. Less consideration should be given to what the other fellow is doing in our own field of work, and more to what is being offered the public in equipment to go with our own stove, refrigerator or kitchen sink. When you try to copy your competitor's successful stove, and make some changes which you consider as being minor, you may completely lose the reason why the other fellow's stove sells so well. At the present time a yellow green is the popular color. Those who are trying to market a green which is so blue as to be almost a robin's egg blue are putting a severe handicap on present sales. Of course, only a small percentage of people are able to tell why the robin's-egg stove does not go with the yellow-green furniture, but there is a subconscious reaction against it."

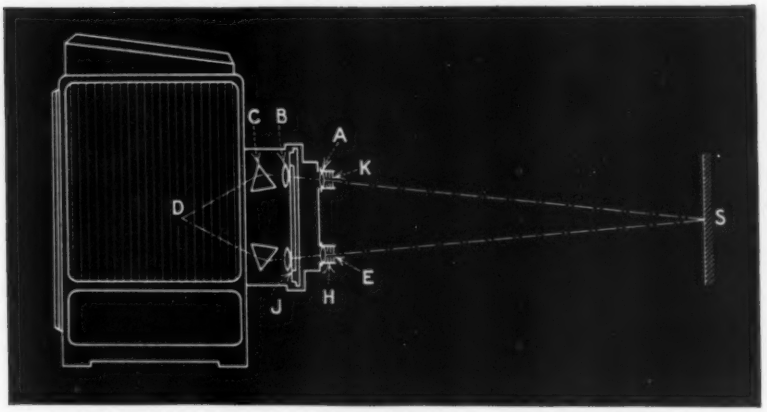


Fig. 1—Diagram of the Mutochrome optical system

COLOR BY OPTICS

DR. IRVING J. SAXL
Consulting Physicist

The importance of color in products is now axiomatic. But how is this selection of colors actually to be done in the most effective way? How can the best definitive result be obtained with the most simple, inexpensive and practical means?

In some instances the designer will be able to predict the effect that a certain color combination will have upon the appearance and the salability of his product. In most instances, however, he is forced to cut and try with cartoons or actual models. And with all the work and time and expense involved in cutting and trying, the experimenter can secure only a limited number of all the actual color combinations which would have been possible.

Now from the laboratories of the physicist comes an apparatus which gives the designer full power to try out any desired color or color combination. With the Mutochrome, the patent rights of which are owned by Adam Hilger, Ltd., London, England, it is possible to vary at the different parts of a product the color of predetermined designs in all three dimensions of color standards—hue, value and chroma. It is possible to change the shade of the color, varying it—at the same time—in its degree of luminosity so that the exact shade of the color desired can be predetermined directly.

Fig. 1 shows a diagram of the optical system of this apparatus. A light source *D* illuminates, through glass prisms *C* and lenses *B*, two different portions of a single lantern slide *J*, whereby the path of the light rays is so directed that they are united upon the same point *S* of the projection-screen.

In the path of the light rays is brought a colored glass *E* at the lower and a differently colored light-filter *K* at the upper part of the diagram. Behind the colored glasses are two separate Iris diaphragms *A* and *H*.

The two pictures from the point *A* and from the point *H* will therefore superimpose on the screen. If therefore one part of a design is photographed upon the upper part of the lantern slide and the other part is copied from the lower part of the same slide, these two parts together will appear in one single united picture upon the screen *S* as though they were of only one projection slide. If the upper part has the fenders of an automobile imprinted on it and the lower one shows the rest of the car without the fenders, the combined projection of both parts will show the car with the fenders at their regular place.

But, as both have been produced upon the united projection through two different channels of light, it is possible to modulate these channels separately and thus to arrange various color combinations independently of each other. For instance, it is possible to insert a red filter into the upper part of the optic. Then the fenders of the car will appear red. The lower part may receive a brown filter, *E*. Thus the combined picture would show a brown car with red fenders.

It is now possible to insert colors other than the two mentioned above. For instance, it is possible to take different shades of blue for the fenders and to combine them with various colors of the body until one is found that blends best within the general color scheme. In addition, by manipulating the iris diaphragms *A*

▼

How can the designer select colors in the most effective way? Dr. Saxl, who has been connected with the Museum of Science and Industry in the City of New York in the preparation of the Exhibition on Color, reveals an optical method

and *H* it is possible to give more or less light to the specific color, thus varying continuously the grey content of the projected color.

Twenty-five Colors

Naturally a much greater number of details can be incorporated in one lantern slide. The Mutochrome shown in Fig. 2 is a 25-color instrument, but 17-, 10- and 7-color instruments are available. Thus a maximum of 25 different colors may be given to 25 different elements of a single design. For cooling purposes the housing for the projection lamp is ribbed and finished in black. Since each optical head, or objective, is provided with a light-tight cap any desired number of colors, from 1 to 25, may be experimented with at a given time. With the instrument comes a large assortment of colored glass disks. One or more of these can be held in front of each objective by means of spring clips.

In the lantern slide, designs are arranged so that each fits exactly before its proper objective. In addition, the same design may be used again at another part of the slide before a different objective. It is thus possible not only to subtract colors from each other by putting various filters in one and the same front opening of the objectives but to add colors to each other by illuminating the same part with 2 or more different colors. It must not be forgotten that all these colors are added to each other in the latter case, not subtracted from each other as in the usual printing processes.

In all Mutochromes, where it is desired to reproduce the color composition selected upon the actual material to be colored, certain difficulties may be encountered because of the fact that the actual colors have to be seen under daylight while the blending of the Mutochrome colors takes place in the dark. For overcoming this difficulty, the color comparator has been devised, the principle of which is shown in Fig. 3.

Direct Comparison

While the light of the Mutochrome is, for instance, in the form of a circle *R*, the center of the circle is in the shadow of an oval mirror *M* so that a round spot *S* is produced upon the actual material which does not receive light from the Mutochrome. But this spot is illuminated by daylight or the equivalent of it (daylight lamp). Thus the color of the specimen, seen under daylight, can be compared directly with the color obtained with the Mutochrome process. The color of the sample (about one-half inch square) is surrounded by the colored ring projected by the apparatus.

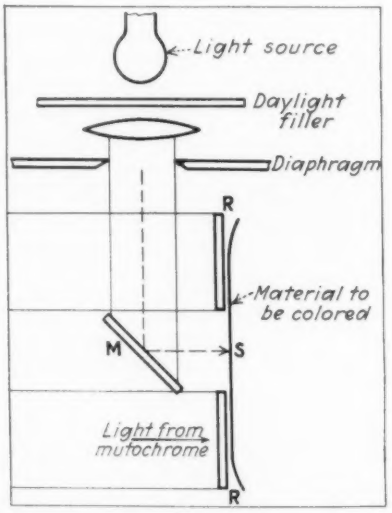


Fig. 3—Diagram of the color comparator optical system

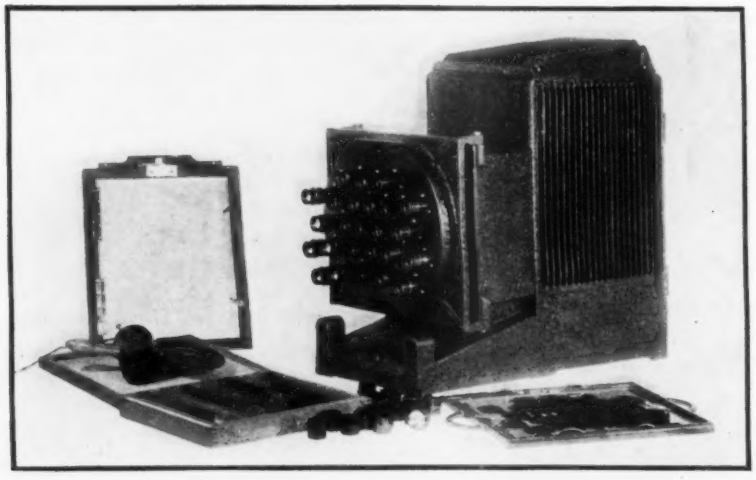


Fig. 2—Mutochrome together with color comparator, color disks, lens caps and slide

The actual operation of the Mutochrome is a simple one. First, photographs of the different parts of the object to be projected are taken or tracings are made upon transparent tracing cloth with ink made from poster black applied to the glossy side. The different photographs or tracings can be photographed together upon one plate, or, in more simple form, they can be attached mechanically upon the slide.

With the projection lamp and the distance between the screen and the projection head adjusted, the slide may be inserted and if necessary moved slightly so that the various designs upon the screen fit exactly into each other. Then various colored filters are put before the objectives, thus coloring the specific part which is projected through them. The brightness of these colors may be adjusted by varying the iris behind each individual color holder.

Color Variation Simple

As no care has to be taken that the color fills exclusively the desired space and no other, the entire effort of the designer can be concentrated upon the color scheme as such. This variation of colors is so simple and goes so quickly that it would even be possible to have a buyer select from such a projection the colors he wants, exactly to his taste.

After the color combination has been fixed, the color comparator comes into action. All colors but one are blended out by covering the other objectives with the caps. This one is matched exactly on a master color card or on the actual material constituting the product. After this color has been determined, the others are matched one by one until every part of the design has received its proper color identification.

Until now the use of the Mutochrome has been confined almost entirely to Europe. But it is safe to say that American engineers and designers will be quick to see its application to their problems. Any device that will improve and simplify practical color design promises to become a valuable asset.

Tooth Contact in Spur and Helical Gears

H. E. KITCHEN

In straight spur or helical involute gears, tooth contact is along a line or lines, depending upon whether one or more pairs of teeth are in contact simultaneously. Tooth wear increases with the intensity of pressure, and this latter depends upon the total length of the lines of contact. It is therefore important to analyze the contact condition.

In standard involute spur gears, the tooth profile can be considered as generated by the unwinding of a strip of paper off the base cylinder, as illustrated in Fig. 1 at *A*. The width of the sheet of paper is equal to the face width of the gear tooth, the unwinding edge being parallel to the axis. The profile of a double-helical gear tooth can be produced in a similar manner, but with the unwinding edge cut in the form of a Vee, either regular or inverted, as shown at *B* in the figure.

The end view in Fig. 2 may be that of either a spur gear or a helical gear of the same axial pitch. The full line shows tooth contact beginning at point *1*, where the line of pressure intersects the base circle, and terminating at point *4*, where the line of pressure intersects the addendum circle. The position of the gear teeth when tooth contact ceases can then be readily shown. It is evident from the figure that during the interval in passing from position *1* to position *2*, there will be two lines of contact, while during the interval in passing from point *2* to point *3* there will be only one pair of teeth in contact. Therefore, the length of line of contact will alternate between a length equal to the face width of the spur gear and a length equal to twice this.

For a pair of single or double-helical gears with the same axial pitch, it is evident from Fig. 2 that the total length of contact is not determined by the product of the number of teeth in contact, and the face width. In the first place, if a straight edge be applied to the working face of the helical tooth, from the root of the tooth at one end to the tip at the other, it will be found that a perfectly straight line exists across this greatly warped surface. It is along such lines that tooth contact occurs.

To find the total length of contact, it is necessary to know the limit of possible contact, as indicated in Fig. 2, and the angle θ and finally the base circle pitch. This length can be found graphically, as shown in Fig. 2, or mathematically.

The mathematical procedure can best be understood by reference to Fig. 3, *A* and *B*.

As an illustrative example, consider a pair of double-helical gears with 16 and 80 teeth respectively, 2 DP, and center distance 24 in. The face width will be taken as 11/DP, or $5\frac{1}{2}$ in. Addendum will be 0.8/DP, or 0.4 in. A 20 deg. pressure angle will be assumed. With both pinion and gear having equal addenda, we have:

$$\text{O.D. of Pinion } (OD_P) = \frac{16 + 1.6}{2} = 8.8 \text{ in.}$$

$$\text{O.D. of Gear } (OD_G) = \frac{80 + 1.6}{2} = 40.8 \text{ in.}$$

Substituting the known quantities in the equations as given Fig. 3 at *B*, we have:

$$EF = 2.287 \text{ in.}$$
$$BC = 7.933 \text{ in.}$$
$$BC + EF = 10.220 \text{ in.}$$

Total length of tangent line, according to Fig. 3A, will be 24 sin 20 or 8.208 in.

Limit of possible contact will therefore be, as shown by Fig. 3, 10.220—8.208 or 2.012 in.

Two helix angles are in general use for double-helical gears, namely, 30 deg. and 23 deg. For single-helical gears $7\frac{1}{2}$ deg. is commonly used. These are the angles on the pitch circle, but as indicated in Fig. 1, the angle θ is measured on the base circle.

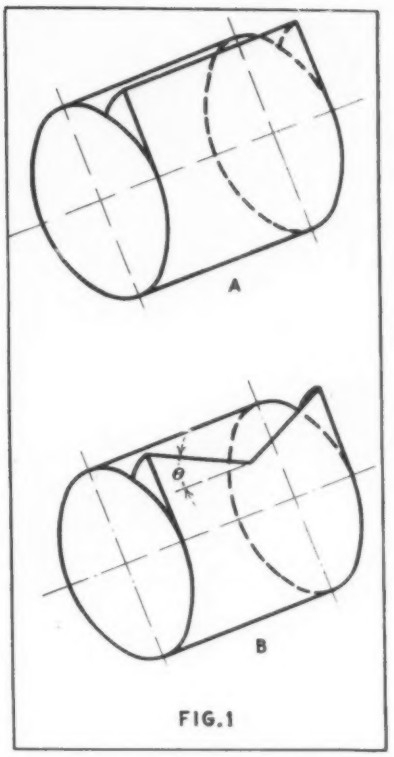


FIG. 1

The faces of involute gear teeth can be generated by unwinding a strip of paper off the base cylinder

In any gear train, rate of wear increases rapidly with the intensity of contact pressure, and the latter decreases with increasing total contact. The author gives a simple method for determining the length of tooth contact for different types of gears, giving quick means for comparing their life on the basis of wear conditions

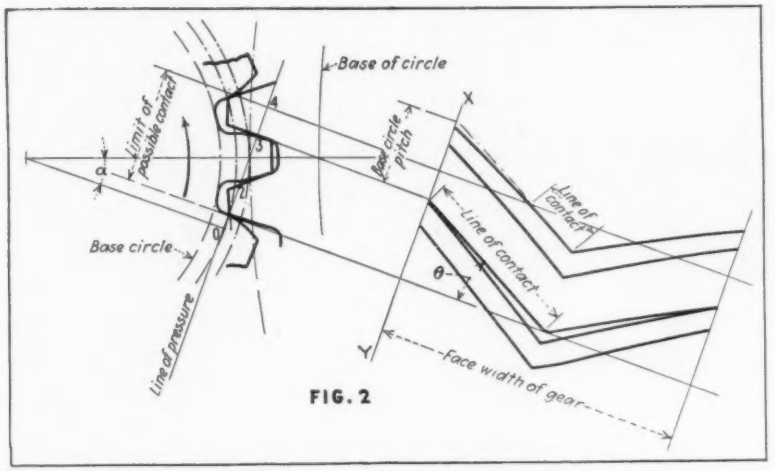


FIG. 2

Showing the limit of possible contact and the lines of contact between the teeth of a pair of helical gears

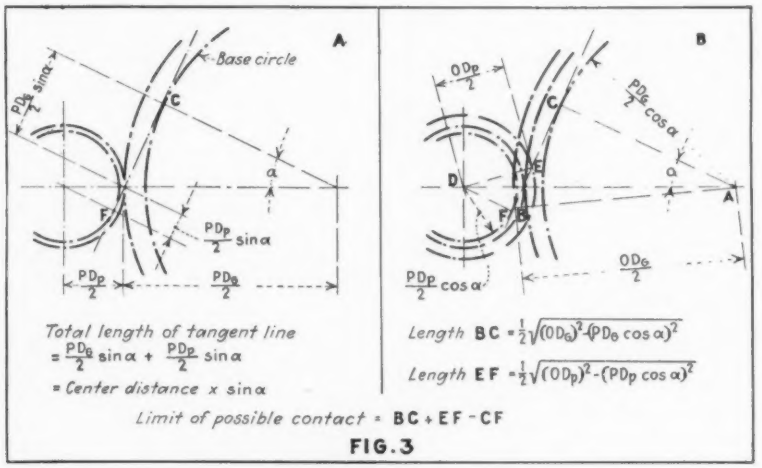


FIG. 3

The equation for the limit of possible contact is derived here, directly from the figure

From the geometry of the figure, we have:

$$\text{Tan } \theta = \frac{\text{Circumference of base circle}}{\text{Lead of helix}}$$

Circum. of base circle = $\pi \times$ pitch dia. $\times$ cos pressure angle α

$$\text{Lead of helix} = \frac{\pi \times \text{pitch dia.}}{\text{Tan of helix angle}}$$

Substituting these values for the circumference of the base circle and the lead of the helix in the above expression for tan θ we have:

$$\text{Tan } \theta = \frac{\pi \times \text{pitch dia.} \times \cos \alpha \times \text{tan helix angle}}{\pi \times \text{pitch diameter}}$$

$$= \cos \alpha \times \text{tan helix angle}$$

Using this equation, we have for the different helix angles measured on the pitch circle the following values of θ:

For 30 deg., θ = 28 deg., 29 min.
For 23 deg., θ = 21 deg., 45 min.
For 7 deg., 30 min., θ = 7 deg., 3 min.

We also have the relation that the base circle pitch equals the circular pitch times cos α or $\frac{\pi \cos \alpha}{DP}$

Thus for 2 DP and a 20 deg. pressure angle, the base circle pitch will be 1.476 in. Expressing the limit of possible contact in base circle pitches, we have for the example under consideration that the limit of contact is equal to the limit of possible contact, 2.012 in., divided by 1.476 in., the base circle pitch. This gives 1.363 pitches of contact, which means that for a pair of spur gears, during 0.363 pitches of rotation there will be two pairs of teeth in contact, while during the remaining 0.637 pitches of rotation there will be only one pair of teeth in contact.

It is now possible to draw the true length of the lines of contact at any instant. Referring to Fig. 2, projecting from the point *3* gives the starting point in Fig. 2 for one line of contact, and by stepping off the base circle pitch along the line *XY*, points for other contact lines are fixed. Angle θ is known, and the useful portions of the lines of contact lie within the area bounded by the limit of possible contact and the face width.

In Fig. 4, are shown different positions of the lines of contact for 30 deg. and 23 deg. double-helical and $7\frac{1}{2}$ deg. single-helical gears respectively. In each figure, view *A* shows the instant when a new tooth is about to engage. View *B* shows by dash lines $1B_1$, $2B_1$, $3B_1$, where contact begins to decrease, and view *C* shows, from the full lines *1C* and *2C*, to the broken lines $1C_1$ and $2C_1$, the period during which contact decreases to the minimum value.

A more direct comparison is offered in Fig. 5 which shows graphically the total length of contact for the four types of gears here considered. The contact conditions are shown superimposed on one chart over a period representing a rotation of two pitches, starting at the point where a new pair of teeth makes contact. Note the abrupt change from maximum to minimum of the spur gears. The 30 deg. double-helical gear is remarkably uniform because of a face width of 11/DP allows tooth contact to overlap. If the face width were increased to 15/DP an equally good showing would be made by the 23-deg. helical gears. It will also be noticed that minimum contact, which of course is of prime importance, occurs when a new tooth is about to engage. Therefore the minimum length can be measured or calculated from view A at the top of Fig. 4.

Analyzing Fig. 4A, we have the following relations:

$$JL = \frac{2\pi \cos \alpha}{DP}$$

$$JK = \frac{F}{2} \tan \theta$$

$$KL = \frac{2\pi \cos \alpha}{DP} - \frac{F}{2} \tan \theta$$

$$QK = QL - KL$$

$$= 2.012 - \left(\frac{2\pi \cos \alpha}{DP} - \frac{F}{2} \tan \theta \right)$$

Length of useful contact on tooth 3, will be:

$$\text{cosec } \theta \times QK$$

$$= \text{cosec } \theta \left[2.012 + \frac{F}{2} \tan \theta - \frac{2\pi \cos \alpha}{DP} \right]$$

Length of useful contact on tooth 2, will be:

$$\text{cosec } \theta \times \frac{\pi \cos \alpha}{DP}$$

The total contact will be the sum of that on tooth 3 and tooth 2 which reduces to:

$$2 \text{ cosec } \theta \left[2.012 + \frac{F}{2} \tan \theta - \frac{\pi \cos \alpha}{DP} \right]$$

Substituting the numerical values, we get:

Total contact = 8.505 in.

For Fig. 5A, the length of useful contact on tooth 2 is $F \sec \theta/2$, and the total contact becomes:

$$2 \left[F \sec \theta - \text{cosec } \theta \left(\frac{2\pi \cos \alpha}{DP} - 2.012 \right) \right]$$

Substituting the numerical values, this gives:

Total contact = 6.768 in.

For Fig. 6A:

Total contact = $F \sec \theta$ = 5.542 in.

Maximum contact is obtained when the apex of the tooth coincides with point B, and by similar procedure, the results are:

30 deg., 8.572 in.
23 deg., 8.814 in.
7½ deg., 10.009 in.

Using the method of analysis presented above, similar calculations can be made for any other designs of spur and helical gears, for the purpose of establishing a basis for comparing unit pressures of contact.

Although in this article the analysis for comparison of wear and tooth pressure has been stressed, this general method has equally important application in problems of gear and shaft vibration resulting from the pulsations and deflections as the load is transferred to the successive teeth. Thus, as in Fig. 5, there is an abrupt change in the length of the contact lines in spur gears, as compared with the almost uniform length of contact line of the 30 deg. double helical gear. Furthermore as the unit tooth-pressures are inversely proportional to the length of contact line, it is apparent that with rapid and abrupt changes in length of contact line the tooth deflections will be affected. This indicates how an analysis such as given above will aid in a study of the vibration problem.

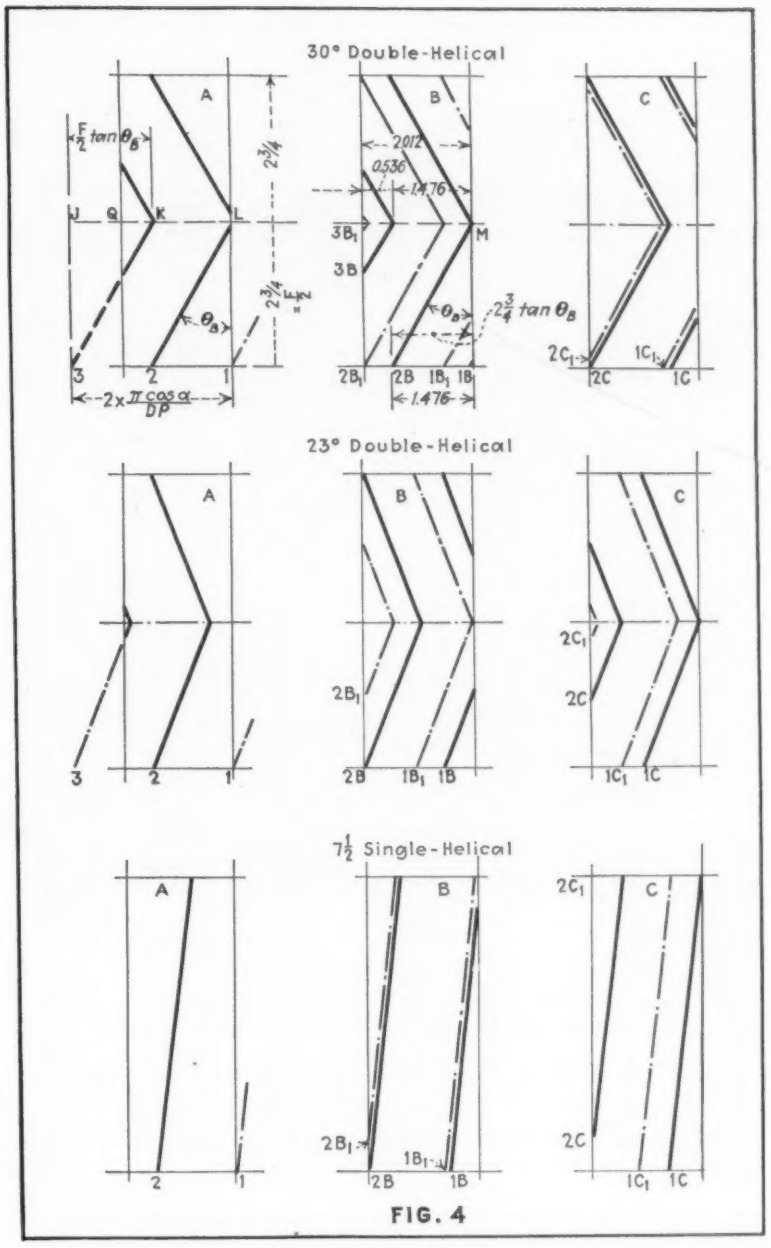


FIG. 4

Different positions of the lines of contact for 30 deg. and 23 deg. double helical and 7½ deg. single helical gears

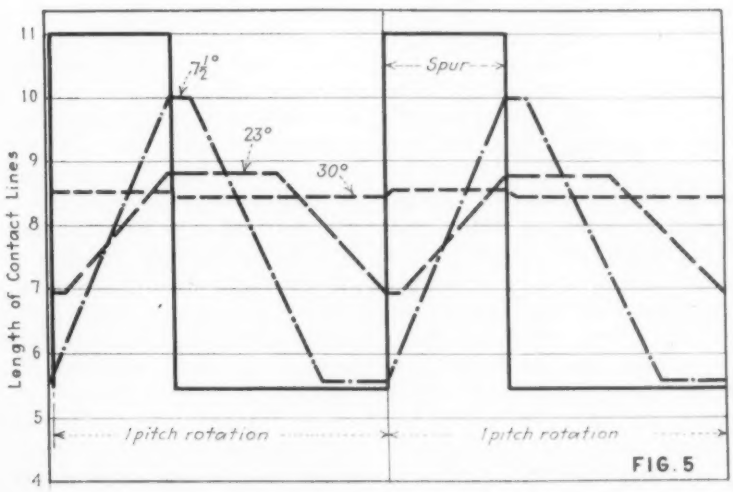
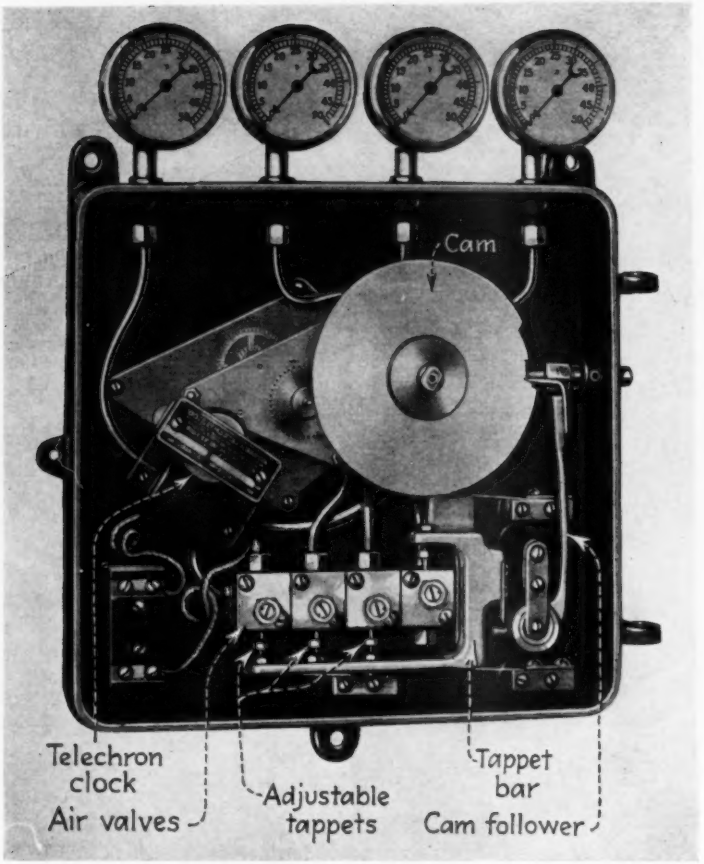


FIG. 5

Graph showing the comparison of the total length of tooth contact in four types of gears

Temperature Control



The mechanism of the *Tycos* time cycle regulator for automatically controlling pressures and temperatures

Automatic temperature, pressure and cycle regulators, formerly purchased by the machine user, are now being applied more frequently by the machinery builders as part of their equipment. This is but a natural development. It enables, the designer of the machine to make special provisions for installing the necessary control units and thus obtain most efficient operation. It also results in compact construction and the elimination of possible changes that might be required in applying the instruments after the machine has been delivered to the user. There is of course also the sales advantage of being able to offer a machine that is complete in all respects.

Temperature and pressure control instruments are essentially the same in the fundamentals of their operation. The only difference is that in the pressure control instruments the movement of the mechanism is a direct result of the fluctuation in pressure, whereas in the temperature control instrument the movement is obtained indirectly. In discussing temperature control, it is to be understood that pressure control instruments are of the same general character. Temperature control equipment consists primarily of a temperature sensitive element which actuates a controlling system and which in turn operates valves or dampers that regulate the flow of the heating or cooling medium. In addition there is sometimes interposed a time cycle regulator for the purpose of maintaining definite temperatures or pressures over given periods of time.

The type of temperature sensitive element employed in industrial machinery and equipment is largely determined by the temperature range and other conditions. For most industrial purposes, expansion type thermometers are used, these consisting of a fluid, an inert gas, or a vapor inclosed in a solid container. Mercury, nitrogen, ether, sulphur dioxide, methyl or ethyl chloride may be employed. The expansion of the fluid, or the increased vapor pressure caused by the rise in temperature, is transmitted through a connecting tube to the mechanism of the thermometer.

Mercury-actuated thermometers of this type can be used for temperatures between —40 deg. F. and 1,000 deg. F. But the length of connecting tube between the bulb and the mechanism must not exceed 50 ft. if compensation for temperature of the connecting tube is to be avoided. For greater distances, up to 500 ft., vapor tension or gas thermometers are generally used.

Vapor tension thermometers can be applied for temperatures between 20 and 750 deg. F. However, in the lower ranges the temperature scale is compressed while in the upper ranges it spreads. This last is an advantage if higher temperatures are being measured. In some instruments this variation in scale spacing is compensated by means of a cam or a spring.

One of the disadvantages of the inert gas expansion thermometer is the larger temperature-sensitive bulb required, which is a factor only where limited space is available. On the other hand, the gas thermometer gives a uniform scale throughout, and can be used for temperatures from —40 deg. F. to 1,000 deg. F., and is good for remote indications up to 500 ft.

Another commonly used expansion type thermometer is the familiar bimetal strip. Two strips of metal, such as copper and manganin, are fastened face to face for their entire length. Any change in temperature will cause the strip to bend by virtue of the difference in the co-efficient of expansion of the two metals. These

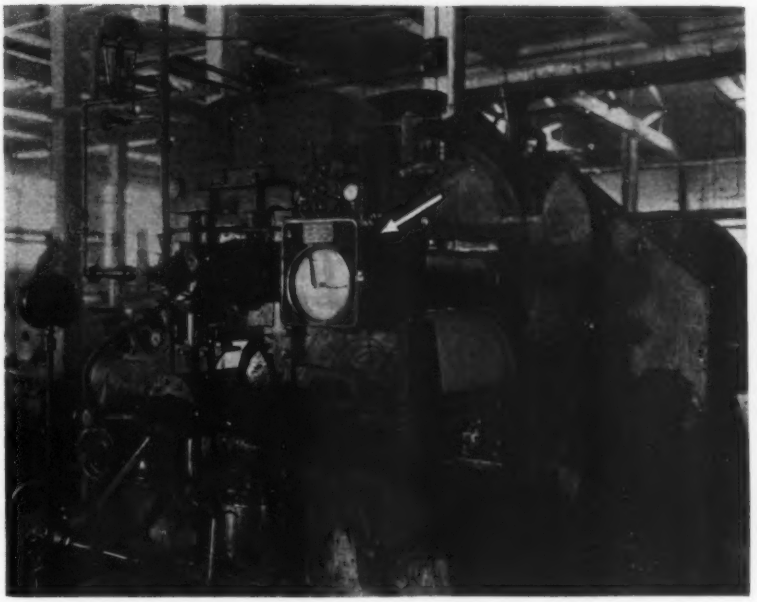
and Pressure Equipment

Their operation and application to industrial process machines and equipment

bimetallic thermometers can be equipped with indicating or recording mechanisms and electric contacts, and can be used to actuate air or water-operated controllers. They have the advantage of simplicity but are not suitable for remote indication. The complete bi-metal unit with controlling contacts or valves is usually placed directly in the zone whose temperature is to be indicated or controlled.

The indicating mechanism of the fluid expansion thermometer is usually a bellows of the Sylphon type or a curved tube similar to the common Bourdon tube used in pressure gages. Sometimes to gain length of movement the tube is made in the form of a helix or spiral. In all cases, the thermometer mechanism may simply indicate the temperature on a dial or may record the temperature on a moving chart driven by clock work or electric motor. In addition, the mechanism for the purposes of temperature control might actuate electric contacts or a pilot valve which in turn controls the operation of the main valve either electrically or by fluid pressure. With vapor tension thermometers, the changing pressure of the vapor is sometimes used to operate the main valves directly through the medium of a flexible member or metal bellows. This direct method of operation is neither as sensitive nor as accurate as the pilot-valve controlled system and should not be applied where accuracy is required.

In planning the installation of a temperature control system, the problem presented is one that involves all of the elements of the behavior of heat. It is not merely a matter of inserting a temperature-sensitive element in a convenient place and the installation of the necessary control valves. Of course the machine or equipment must be designed so that the desired heat and temperature distribution will be obtained. Then, for the best location of the temperature-sensitive element, the temperature zone which responds most quickly to the changes in the operating temperature of the machine must be located. One reason for the sometimes unsuccessful installations of temperature control systems is the failure of the thermometer to register the desired true mean temperature.



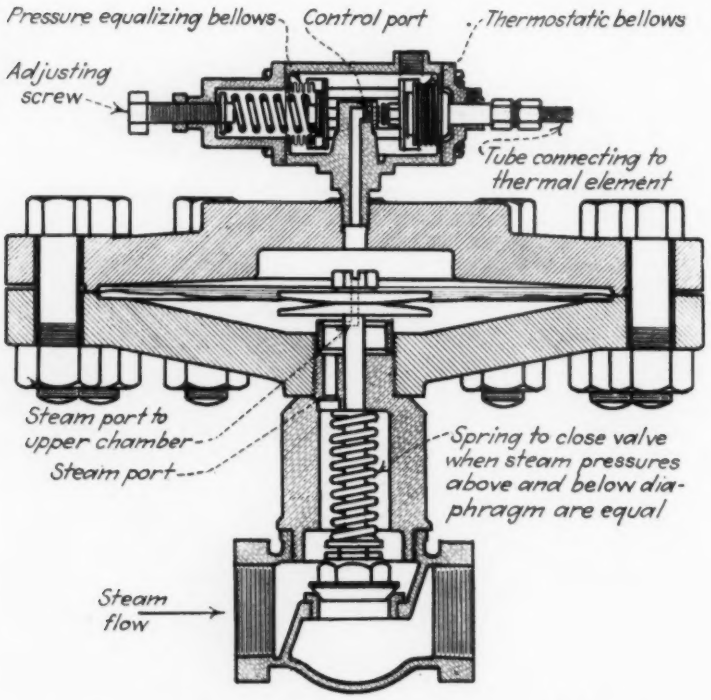
Automatic temperature control applied to textile singeing machine. The arrow points to the control instrument and temperature recorder. Note the connecting tube from the instrument to the bulb and the diaphragm valve in the upper left corner of the picture

Another factor that must be considered in the design of an automatic temperature control system is temperature lag. This is frequently inherent in the design of the machine or equipment. In so far as temperature control is concerned, the effect can be minimized by temperature sensitive elements that respond quickly and by placing them in a zone where they will be subjected immediately to any fluctuations in the the designed operating temperature.

The temperature controlled valves that regulate the flow of the heating or cooling medium are, mechanically, either direct-acting or reverse-acting. The direct-acting valves open with increased pressure on the diaphragm; the reversing-acting valves are closed thereby. The valves may be of the throttling type or of the open-and-shut type. They may be actuated by a flexible diaphragm or bellows, using air, steam, or water for the operating medium. For electrical control, the valves are solenoid or motor operated.

In addition to maintaining a definite temperature or pressure, temperature control equipment can, by means of a time cycle regulator be used to operate the machine automatically throughout any desired time-temperature or time-pressure cycle. Such a regulator is shown in the accompanying illustration. It is merely a matter of cutting the cam to procure any desired time-temperature or time-pressure cycle.

The "Tycos" time-cycle regulator shown here has four air valves, although any smaller number can be used as required. Essentially, the instrument consists of a ball bearing


Pressure equalizing bellows
Control port
Thermostatic bellows
Adjusting screw
Tube connecting to thermal element
Steam port to upper chamber
Steam port
Spring to close valve when steam pressures above and below diaphragm are equal
Steam flow

Pressure control head built by C. J. Tagliabue Manufacturing Company. Opening of the control port, regulated by the thermostatic bellows causes pressure drop in the upper chamber of the diaphragm. The valve opens against the spring pressure

mounted cam driven by a Warren Telechron clock, and a cam follower that actuates a tappet bar on which adjustable tappets are mounted for operating the air valves. The desired setting of the valves is accomplished by adjusting the tappets. By using different motor speeds and gear train ratios for driving the cam, the period of its revolution may be made anywhere from 5 min. to 40 hours. Within even comparatively wide limits, the speed of the cam can be adtered by changing only two gears.

Time-temperature cycle regulators have three general types of application: (1) for continuous processes where certain temperatures or pressures are to be held periodically in a given order over an extended period of time; (2) for processes where certain temperatures or pressures are to be maintained over periods of time, but the succeeding duplicate cycle is to be started manually; (3) processes wherein the final holding period must be adjustable.

Problems inherent in a complete temperature control system, are illustrated in the design of the "Buflovak" vulcanizer manufactured by the Buffalo Foundry & Machine Company. The primary object of the system is to provide a uniform cure of rubber goods by a uniform distribution of heat throughout the vulcanizer. If there is a wide temperature variation over-curing is apt to occur at the higher temperature, which materially impairs the wearing qualities of the rubber.

The vulcanizer is built along conventional lines, with sheet steel tank and quick opening doors. The main features are the arrangement of the coils and the circulating system.

In operating this equipment, with the material loaded and the door closed, air admitted by a hand-controlled valve builds up the desired pressure, usually about 30 lb. per sq.in. Through a pressure regulator, the air then enters a Tycos time-cycle regulator such as illustrated. When the desired pressure has been reached, as indicated by a signal light controlled by a pressure switch, the hand-controlled air valve is shut off, the air pressure thereafter being regulated by an automatically controlled diaphragm valve.

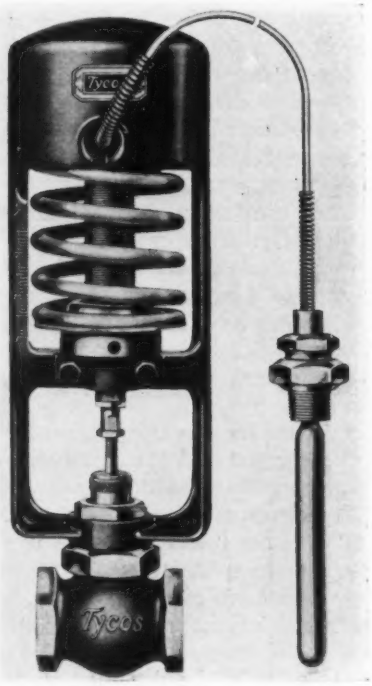
At the same time, one of the air valves of the time-cycle regulator opens a reverse acting. open-and-shut type diaphragm valve in the steam line to the heater coils. This is for preventing admission of steam when no pressure is in the vulcanizer.

The time-temperature regulator is adjusted to provide the raise in temperature and length of cure desired. One of the air valves of the regulator controls a diaphragm valve that throttles the steam supplied to the heating coils, thereby maintaining a definite steam pressure, as controlled by the temperature in the vulcanizer through a temperature-sensitive bulb in the vulcanizer. Thus the correct amount of steam at the proper pressure is admitted automatically.

Forced circulation within the vulcanizer maintains a uniform temperature throughout, no air being taken in from the outside. A swirling action is obtained by baffling, to prevent stratification and dead air pockets.

Condensation from all the coils is connected into one line and discharged through a diaphragm valve which is also controlled automatically.

Application of automatic temperature and pressure control instruments in connection with time-cycle regulators to industrial machinery and equipment is rapidly being extended, particularly in the manufacture of individual vulcanizers for tires and tubes. Time-temperature cycle equipment has also been applied to canning retorts, heat-treating furnaces and coffee roasters, and can be applied to any machine or equipment that operates on a time-temperature or time-pressure cycle.



Self-acting temperature regulator of the Taylor Instrument Companies, wherein a double seamless bellows operates the valve directly.

Time—Torque—Acceleration in Centrifugal Clutches

L. J. McDONOUGH and E. NIEDERER

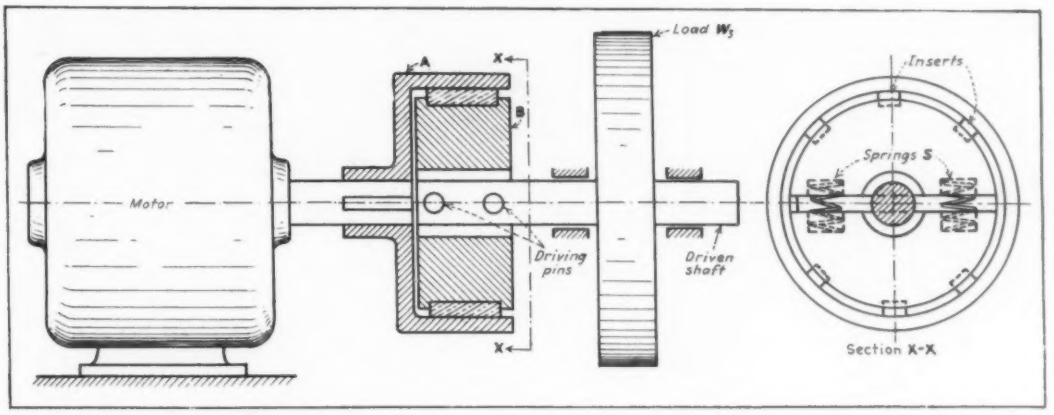


Fig. 1—Design of a typical centrifugal clutch. The inner member *B* is split and the halves are thrown radially outwards

A need often arises in machine design for a device by which the time of acceleration of a load can be widely controlled. Fig. 1 is a diagrammatic sketch of a centrifugal clutch which was successfully used to meet this requirement. The problem that presents itself to the engineer designing such equipment resolves itself into the determination of the time required for acceleration and the maximum accelerating torque, for the purpose of proper proportioning of the component parts.

In the following, a general equation has been developed with which it is possible to compute the time required to accelerate a given load to a desired speed. A specific numerical problem has also been solved.

Part *A* of the clutch is connected directly to the driving motor. The inner part *B* is connected to the load and is split so that its parts can move radially. Springs *S* provide sufficient pressure between *A* and *B* to initiate rotation of *B*. Due to this initial rotation, centrifugal forces act upon *B* creating a greater pressure between *A* and *B*. This pressure increase varies as the square of the speed of *B*, which results in a progressive increase in accelerating torque. Inserts of wood are provided in *B* so that the slippage will be between *A* and the inserts. This provision is also necessary in order to secure a high coefficient of friction.

The following is the analysis of the mechanics of such a clutch, which is necessary as a basis for its design.

In Fig. 1, assume the following:

R = inside radius of member *A* in ft.
W_1 and r_1 = weight in lb. and radius of gyration in ft. of *B*
W_2 and r_2 = weight in lb. and radius of gyration in ft. of shaft.
W_3 and r_3 = weight in lb. and radius of gyration in ft. of load.
ω = angular velocity of *B* in radians per sec.
α = angular acceleration of *B* in radians per sec. per sec.
P = spring pressure or total pressure in lb. between *A* and *B*.
μ = coefficient of friction between *A* and the inserts in *B*.
T = Total torque transmitted by the clutch, in ft. lb.
t = time in seconds.
I = total moment of inertia of the driven system.

The torque transmitted, when running, resulting from the action of the springs is $P\ \mu\ R$ ft. lb. The torque transmitted resulting from the centrifugal force of *B* on *A* is

$$\frac{W_1\, r_1\, \omega^2\, \mu\, R}{g} \text{ ft. lb.}$$

Hence:

$$T = P\ \mu\ R + \frac{W_1\, r_1\, \omega^2\, \mu\, R}{g}$$

But

$$T = I\alpha$$

$$I\alpha = P\ \mu\ R + \frac{W_1\, r_1\, \omega^2\, \mu\, R}{g}$$

$$\alpha = \frac{P\ \mu\ R\ g}{W_1 r_1^2 + W_2 r_2^2 + W_3 r_3^2} + \frac{W_1\, r_1\, \omega^2\, \mu\, R}{W_1 r_1^2 + W_2 r_2^2 + W_3 r_3^2}$$

Let

$$C = \frac{P\ \mu\ R\ g}{W_1 r_1^2 + W_2 r_2^2 + W_3 r_3^2}$$

$$K = \frac{W_1\, r_1\, \mu\, R}{W_1 r_1^2 + W_2 r_2^2 + W_3 r_3^2}$$

$$\alpha = K\,\omega^2 + C = \frac{d\omega}{dt}$$

$$dt = \frac{d\,\omega}{K\,\omega^2 + C},$$

$$t = \int \frac{d\,\omega}{K\,\omega^2 + C} = \frac{1}{\sqrt{K\,C}} \arctan \omega \sqrt{\frac{K}{C}}$$

As an example, find the times required to accelerate a certain clutch

to various speeds up to 1,800 r.p.m. using the following assumed values:

$W_1 = 3$ lb., $r_1 = 1\frac{1}{2}$ in., $R = 2$ in.
$W_2 = 5$ lb., $r_2 = \frac{1}{4}$ in., $P = 5$ lb.
$W_3 = 40$ lb., $r_3 = 4$ in., $\mu = 0.3$

Applying the above equations:

$$C = \frac{5 \times 0.3 \times 0.1666 \times 32.2 \times 144}{(3 \times 2\frac{1}{4}) + (5 \times \frac{1}{4}) + (40 \times 16)}$$

$$= \frac{1159.2}{648} = 1.7888$$

$$K = \frac{(3 \times 0.125 \times 0.3 \times 0.1666)\,144}{648}$$

$$= \frac{0.01875 \times 144}{648} = 0.0041666$$

At a speed of 1,800 r.p.m.,

$$t = \frac{1}{\sqrt{1.7888 \times 0.0041666}} \times \text{arctan } 188.5\sqrt{\frac{0.0041666}{1.7888}} \quad (1)$$

$$= 11.598 \text{ arctan } 188.5 \times 0.04826$$
$$= 11.598 \text{ arctan } 9.097$$

Arctan 9.097, equals 83.733 deg., which must be divided by 57.2 to change to radians. Hence:

(for 1800 r.p.m.)

$$= \frac{11.598 \times 83.733}{57.2} = 16.96 \text{ sec.}$$

By making the arithmetical computations in tabulated form, the work can be shortened considerably. The values of K and C are constant, the only variable being the selected values of ω, which greatly simplifies the calculations. The results for the value of t for different speeds are given in the accompanying table.

To calculate the torque in ft.-lb. transmitted at various speeds, we have:

$$T = P\,\mu\,R + \frac{W_1 V_1^2}{g\,r_1}\,\mu\,R$$

V_1 = Linear velocity at rad. of gyration r_1 of weight W_1 in ft. per sec.

At 10 r.p.m., all units being in ft. or lb., we have:

$$\frac{W_1 V_1^2\,\mu\,R}{g\,r_1} = 0.000638 \text{ ft. lb.}$$

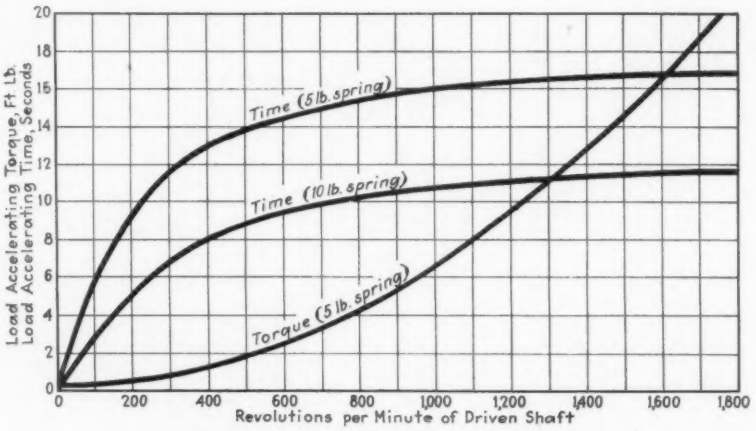


Fig. 3—Effect of varying spring pressure or moment of inertia, on the time for acceleration

Speed, Time, and Torque for a Typical Centrifugal Clutch

R.p.m.	t	T
1800	16.96	20.95
1700	16.88	18.70
1600	16.80	16.60
1400	16.60	12.75
1200	16.35	9.43
1000	15.98	6.63
800	15.40	4.33
600	14.33	2.55
400	12.90	1.27
300	11.49	0.825
200	9.17	0.506
100	5.43	0.314
50	2.87	0.266

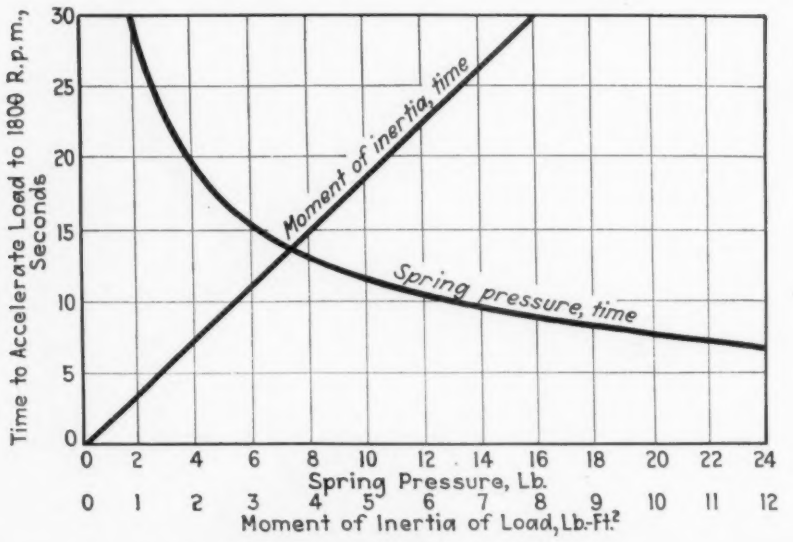


Fig. 2—Time-speed and torque-speed curves

Substituting the proper values:

$P\,\mu\,R = 0.25$ ft. lb.

Hence

$T = 0.25 + 0.000638$

After having determined the torque T for 10 r.p.m., the torques corresponding to any other r.p.m. can be determined by simply multiplying 0.000638 by $(\text{r.p.m.}/10)^2$ where r.p.m. represents the number of revolutions for which the torque T must be determined, and then adding 0.25. The results for this example are shown in the table.

In Fig. 3 is shown the time-speed relations for the above example for both 5 and 10 lb. of spring pressure, and the torque-speed relations for the 5 lb. spring pressure. The curves in Fig. 2 illustrate the effect of varying either spring pressures or the moment of inertia of the load.

It can be seen from curve in Fig. 3, that in the low ranges of spring pressure a slight change in pressure has a great effect upon accelerating time. In using spring pressures of large magnitude, however, the same amount of change has relatively little effect. The curves illustrate that time for acceleration varies directly as the moment of inertia of the load. An inspection of the equation for t indicates that this is true only for those cases in which the moment of inertia of the clutch is relatively small.

COMMUNICATION AND COMMENT

Magnetically Operated Shear

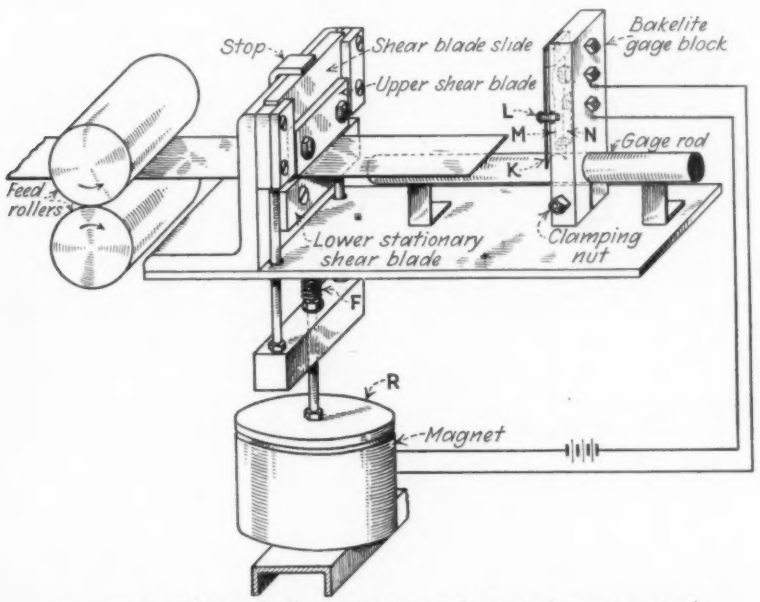
WALTER B. DAHM

The accompanying drawing shows an automatic shear attachment which has been in operation for a period of several months. Its simple construction and ease of adjustment should make it readily adaptable to many operations of similar nature.

The machine for which this attachment was designed performs a special operation on strip metal, after which it is necessary to cut the strip into pieces of various lengths. Sometimes as few as one hundred pieces of a certain length are required, following which several thousand pieces, probably two or three times as long, are needed. It is desirable to be able to change the cut lengths without shutting down the machine.

In operation, the metal strip is pulled through the machine at a constant rate of speed by the feed rollers. It then passes between the upper and lower shear blades, the upper blade being mounted on a slide which is normally held up against the stop by a spring F. A Bakelite gage block is slidably mounted on the gage rod, being held in the desired position by the clamping nut.

As the metal strip moves through the shear its end finally pushes against the spring brass contactor K. A small adjustment screw L in this brass strip bears against upper contact M causing it to close an electric circuit through lower contact N. Current then flows through the winding of the magnet. This pulls down steel plate R and with it the upper shear blade, thereby shearing off the portion of metal strip between the shear and strip K. The metal strip drops to the machine table thereby releasing pressure on K and opening the electric circuit. Spring F then returns the shear blade to its upper position.



Automatic and practically instantaneous in its action, no intermittent drive to stop the strip during the shearing stroke is required on this magnetically operated shear

This entire action occurs so rapidly that the feed rolls can continue to revolve, thereby eliminating the need of an intermittent drive mechanism. The small movement of the strip through the feed rolls while the shear is closed is taken care of by slight bulging or bending of the strip between the rolls and the shear.

Commercial magnets or solenoids, for either a.c. or d.c., can be purchased as standard units. If d.c. magnets are to be operated off of a.c. supply, a dry plate rectifier can be used for converting the a.c. current.

The alert special-machine designer will, of course, immediately see other applications of this or a similar device for punching holes at regular intervals or for code dating parts or strips that are being fed through rolls.

Solving Higher Degree Equations

NORMAN M. WICKSTRAND

Higher degree equations are apt to contain trigonometric, inverse trigonometric, logarithmic or exponential terms. These conditions increase the difficulty of solving by cut and try methods and might render graphical methods, such as the one given by E. N. Kemler in the March number of *Product Engineering*, of little use. In such instances, Newton's approximation rule will be found of considerable value. It can be found with full explanation and theory in "Higher Mathematics" by Merriman and Woodward. In this article, only the method of the practical application of this theory, without proofs, will be given.

In using this method, a root approximately correct to the first significant figure is found by trial. From the nature of the problem, this can usually be found at once, merely using one's judgment. Let x_1 equal this first approximate root. The second approximate root, $x_2 = x_1 - \frac{f(x_1)}{f'(x_1)}$, where $f(x_1)$ equals the value of the equation with x_1 substitute for the unknown x and $f'(x_1)$ equals the first derivative of this equation with x_1 substituted in place of x. A second

substitution in the form of $x_3 = x_2 - \frac{f(x_2)}{f'(x_2)}$ will usually give sufficient accuracy.

Using a table of squares and cubes the amount of work involved is not as great as might be presumed. The advantage of this method over a cut and try or Horner's method is that in each successive try a result definitely nearer to the exact root is found and frequently each successive try will give one more significant figure in the root and sometimes two.

There is, however, one limitation to the method. If a curve is plotted of the remainders for different values of x and a maximum or a minimum occurs between x_1 and the real root, the method fails. Such a condition will be made plainly evident in that successive values of $f(x_1)$ will not be progressively decreasing. Such situations, however, are so rare that one need not hesitate using this method.

To illustrate the method by a simple algebraic cubic:

$$3x^3 + 4x^2 - 3x - 2 = 0$$

To obtain the first root for this method it is usually most convenient, to try some power of 10 which gives a satisfactory trial root. In this example we use 1 as the trial root, that is, 10 to the exponent 0 which equals 1. The remainder is found to be 2. The continuation of the process is shown in the accompanying table.

As can be seen readily, this gives considerable accuracy with a relatively small amount of work. A second root may be found by dividing the original equation by (x — the root found), in this example, (x — 0.832), and then repeating the above process. In this problem, however, the quadratic formula may be used for obtaining the balance of the root. Furthermore, in practical problems one real root is usually all that is needed and therefore the process seldom has to be repeated.

Tabulated Form of Procedure for Solving Higher Degree Equations

	(1)	(2) $f(x)$	(3) $f'(x)$	(4) $\frac{(2)}{(3)} = \frac{f(x)}{f'(x)}$	(5) New Root (1) — (4)
First root by trial.....	1.000	2.00	14.00	0.143	0.857
Second root.....	0.857	0.26	10.46	0.0249	0.832
Third root.....	0.832	0.007	9.89	0.00007	0.83193
Fourth root.....	0.83193	....	Etc.	.....	.....

Standard Lettering Should Be Abolished

E. Chat. Shanks

A good draftsman or a designer is necessarily an individualist. Men who have no opinions of their own, who fall easily into decreed forms, who meekly adapt themselves to dilatory new styles, who laboriously attempt to suit the whims of each new rule-maker they encounter, belong in some routine clerical position rather than on the drafting board. The men who are constantly expressing themselves in their work, who have no patience with painstaking dogmatism, who fret under the restraint of curbing restrictions in unimportant matters, who grumble over dictatorial prohibitions, are the ones who do the original work on which profits are made for the company.

The humorous side of it is that the very men who resented such practices when they worked on the board, frequently become martinets of minute regulations when they are placed in charge of a drafting room. Why this is so is difficult to determine. If the engineers who make the rules had to work under them, they would be discontented, to say the least.

Perhaps the most aggravating and costly rules are those regarding lettering. Nearly all drafting rooms specify the style to be used: single stroke or down stroke; all capitals or capitals and lower case; vertical, back-hand or slant; some even specify a certain angle of slant. Frequently the height of the letters is decreed in thirty-seconds, or even in sixty-fourths, of an inch and extremists have gone so far as to designate the exact width of each letter.

The things which the bigots ignore are: lettering is as individual as writing and a certain style or size of letter may be very difficult and awkward for most men. The only reason for lettering a drawing, instead of writing upon it, is to make it easier to read. Therefore, what possible difference can it make what style or size of lettering is used as long as it is legible? Why spend needless time in making letters to conform to opinion of a rule-maker when the draftsman can do better-appearing, more legible lettering in the style to which he has trained his hand?

An example is found in a company which has two distinct divisions with a common drafting room. The older division uses slant lettering the newer one uses vertical lettering. This is so because the new work was originally started with a single draftsman who was partial to vertical letters and he persuaded the manager it should be used. His argument was that no two men lettered at the same angle and subsequent changes were noticeable if made by different men; that if vertical letters were used they would always look alike. Of course, they do not look alike whether vertical or slant, when made by different draftsmen, but the ruling was made and is still in effect. Since the detailers do work for both divisions, mistakes in style constantly occur and, because ink tracings are used, they frequently must be made over again.

Enforcing a standard style for lettering results in time consuming methods. It is a costly practice and unessential. It limits the individuality of the draftsman, cramps his self-expression, and curtails the benefits of his experience. It is expensive to the employer and adds nothing to the clarity of the drawing.

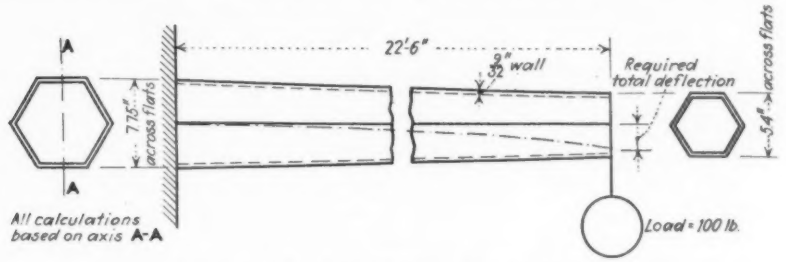
[Editor's Note—*The author raises the question of the necessity, or even the desirability, of a standard style and size of lettering. Such an idea runs counter to the modern tendency toward universal standardization, yet there is evident merit to his arguments. Should drawings reflect the individuality of the draftsman or the more or less arbitrary rules of the company?*]

A Problem in Deflection

L. V. JOHNSON

In the accompanying figure is shown a hollow tapered hexagonal cantilever beam with the larger end fixed and the other end having a load of 100 lb. suspended from it. Neglecting the weight of the beam, what will be the total deflection at the free end?

I have found this problem extremely interesting and will appreciate any solution that might be submitted by mathematically inclined readers.



What will be the deflection at the free end?

Designing for Hollow Pressed Pins

M. V. ZAUSTINSKY

One of the most useful engineering principles is that of equal stresses or equal resistance. It results in economy of material, and when applied to the forced fit, it also gives economy of increased safety and sometimes economy of labor.

The greatest equivalent simple stress in a hollow pin will be its tangential stress at the hole. Using the well known Lamé's formula, we may write:

$$S_t = \frac{E\delta}{b} \cdot \frac{1 - \frac{b^2}{c^2}}{1 - \frac{a^2}{c^2}} \qquad (1)$$

E = the modulus of elasticity
b = the radius of the bore of the hub
c = the outer radius of the hub
a = the radius of the hole in the pin
δ/b = the allowance per in. of nominal diameter of the fit

The greatest equivalent simple stress in the hub, according to the maximum shear theory, will be:

$$S_s = \frac{E\delta}{b} \cdot \frac{1 - \frac{a^2}{b^2}}{1 - \frac{a^2}{c^2}} \qquad (2)$$

These formulas are derived on the assumption that both mating parts are of the same material.

Using the principle of equal stresses, we may write:

$$S_s = S_t$$

Substituting Equations (1) and (2), we will obtain:

$$\frac{b}{c} = \frac{a}{b}$$

We see that in order to satisfy the condition of equal maximum stresses in both parts, the relative thicknesses of the hub and of the pin must be equal.

In Fig. 1 two families of the curves, S_s and S_t are plotted, the abscissas being the relative wall thickness of the hub. Each curve corresponds to a definite ratio of a/b, the relative wall thickness of the bored pin. The point of intersection of two curves for the same relative thickness will be the point of equal stresses in the hub and bore. Such, for example, is point B, in which the stresses for $a/b = 0.7$ are intersecting. If we will proceed to the right of this point, that is, in the direction of decreasing hub wall thicknesses, the stresses in the hub will increase and in the pin will decrease. Thus, the maximum stress of the system will be in the hub. But, if we will proceed to the left point B, the conditions will reverse, and the maximum stresses of the system will be in the pin, as can be determined readily from the curves.

Therefore, point B is the point of minimum stresses in the system. By boring the pin hole larger we decrease the stresses in the hub and increase them in the pin, thus increasing the safety of the hub. It is evident, that in such case the pressure between two mating parts, or the gripping power of the joint will decrease. But in many cases such decrease is permissible. And where it is not permissible, the amount of allowance may be increased in order to produce the desired pressure. Then we have another advantage of bored pins—increased allowance, which makes possible increased tolerances and cheaper cost of production.

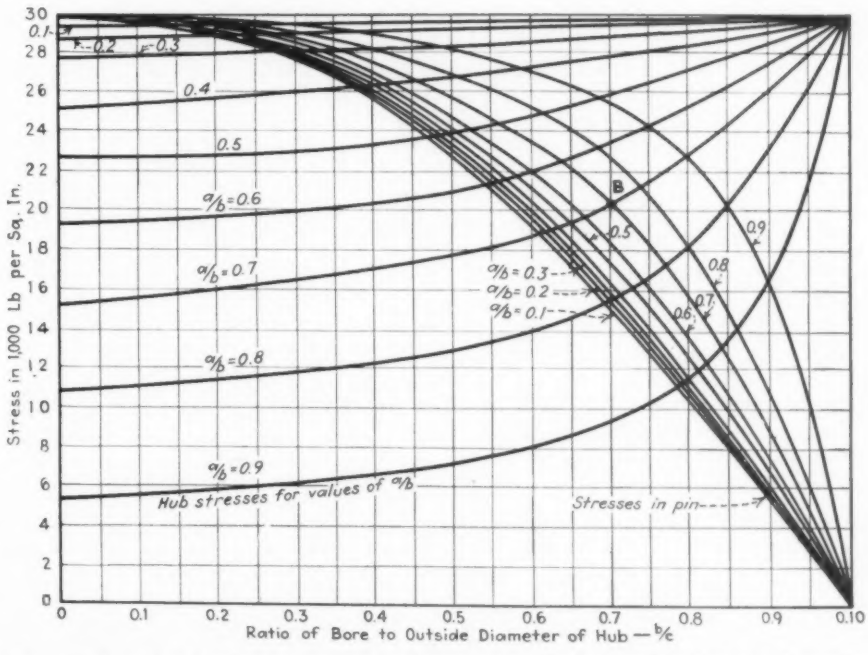


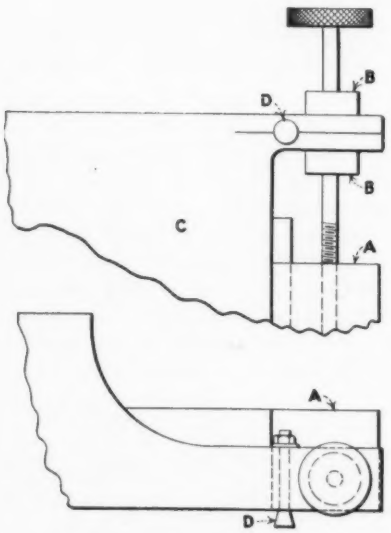
Fig. 1—Chart for determining stresses and proportions for hub and hollow pin diameters

Adjustment Screw Mounting

JOHN E. HYLER

On certain types of machines where accurate maintenance of adjustment is a factor, and where vertically-mounted adjustment screws are to be used, the ordinary type of mounting is a source of considerable trouble. Referring to the accompanying figure, the inevitable wear between the collars *B* and the machine frame creates lost motion, with result that *A* will not move until this lost motion is first taken up in the direction of travel.

A simple method for taking up this wear is shown. The frame extension is split with a kerf and drilled with a hole to receive the bolt *D*, which is fitted with a gradually tapered head. By taking up this bolt with the nut, a sensitive adjustment for taking up the wear is obtained, and one which will not disturb the setscrews or the taper pins, as the case may be, with which the two collars *B* and *B* are mounted. The slitted extension of *C* is simply spread to take up the wear.

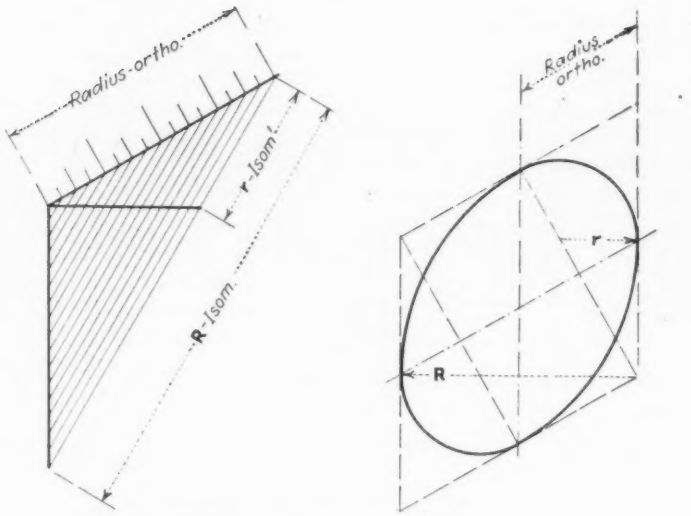


Wear between the collars *B* and *B* is compensated by taking-up on the tapered-head bolt

Scale for Isometric Radii

BY J. HOMEWOOD

In drawing arcs and circles in isometric, the common procedure is to construct rhomboids circumscribing the isometric circle, and thereby obtaining the necessary radii and their centers for constructing the circle.



The radii for constructing circles in isometric can be obtained directly from this chart

Much of this work can be eliminated by using the scale as indicated in the accompanying figure. The scale can be laid out on the drawing being made, or it can be constructed on a triangle for permanent use.

The scale representing the orthographic measurement is laid out on a line making an angle of 30 deg. with the horizontal. The shorter and longer isometric radii are indicated in the figure. To the right is shown the method of locating the center of these radii and drawing the circle.

DEFLECTIONS

MECHANICAL CERAMICS

Where high heat and acid resistance are indicated the ceramics should not be overlooked, particularly in small parts. "Lava," made from powdered steatite and marketed by several companies, can be made up into intricate shapes without expensive molds. It can be cut or threaded before final baking. The finished parts are harder than glass and have high strength. Incidentally, they can be used as excellent insulators.

GRAIN SIZE

According to researches by General Motors engineers, as reported in *Mechanical Engineering*, an improvement of 50 to 400 per cent in machinability of forgings can be obtained by using forgings of maximum density. Grain size in forgings will soon be another item for the engineer to specify.

RUBBER AND BRASS

Where, as in automotive rubber mountings, rubber must have the tightest possible bond with steel, experience has shown that unless certain safeguards are observed vulcanization brings about separation. According to one expert, the steel must be cleaned electrolytically, rather than pickled, and then plated thoroughly with a good grade of 70-30 brass. Engineers will save themselves grief by writing these points into their specifications.

PANIC-PROOF ALARM

An automatic fire detector, instead of setting off a nerve-wracking and panic-producing gong, responds in three ways: It sends an alarm to the fire house, directing the firemen to the side of the building nearest the fire, while in front of the building itself an arrangement of lights shows them in what particular part the fire is located. Green arrows light up, pointing out the correct route for exit. A phonograph begins to play music through loud speakers. Then a calm but commanding voice says: "There is a fire emergency but no immediate danger. You must leave the building . . . . Steady now, don't rush. Follow the green arrows." Then is heard a brass band striking up a martial air, and the sound of approaching fire sirens.

A.S.T.M. Thirty-Fifth Annual Meeting

Many papers of special interest to designing engineers were on the program of the Thirty-Fifth Annual Meeting of the American Society for Testing Materials, held at Atlantic City, June 20-24. The symposium on steel castings was presented at a joint session with the American Foundrymen's Association. At the session on the Effect of Temperature on Metals, the joint research committee of the A.S.M.E. and A.S.T.M. submitted its report.

Perhaps the most significant session of all was the round-table discussion on "Acquisition of Good Data," a misleading title because it is not indicative of the broad scope of the general subject of "Statistical Control," of which it forms a part. In spite of the fact that the subject is relatively new to this country, in that it is not widely known, it was indicated at the meeting that rapid strides are being in the fields to which it might apply.

Statistical control is in effect the scientific establishment of expected qualities of dimensions and characteristics on the basis of past experience. It is used for analyzing the value and correctness of data, the establishment of economic standards of quality, and the scientific control of inspection. Quoting W. A. Shewhart, of the Bell Telephone Laboratories, who is the leading exponent of statistical control in this country, "Economic standards of qualities are standards such that, under the given condition with respect to development of science and development of human wants, there is established a balance between the economic value to the consumer of any possible modification in quality standards and the costs associated therewith. Economic control, as applied to production and design, maintains the economic standards." The papers presented at the round-table discussion were: "Statistical Nature of A.S.T.M. Data," by R. F. Passano, of the American Rolling Mill Co., "Specifications of Standards of Quality," by N. F. Harriman, of the Federal Purchasing Board, "Inspection Specifications," by G. D. Edwards, Bell Telephone Laboratories, "Need for Control in Sampling Inspection," by H. F. Dodge, of the Bell Telephone Laboratories, "Criteria for Rejection of Observations," by P. R. Rider, Washington University, "Methods of Dealing with Discordant Observations," by E. W. Washburn, U. S. Bureau of Standards, "Elimination of Systematic Errors," by H. C. Dickinson, U. S. Bureau of Standards, and "Role of Statistical Method in Economic Industrial Standardization," by W. A. Shewhart, of the Bell Telephone Laboratories.

A more general recognition of the value of this scientific subject would have a tremendous effect. It would undoubtedly result in an entirely different method of interpreting the results of acceptance testing. The present method of acceptance or rejection on the basis of the success or failure of a specified number of samples, would be entirely discarded.

Under a different name, statistical control originated in Germany in 1922. At the plant of Bauer & Chaurte, manufacturers of high quality nuts and bolts, it was desired to establish a more scientific method of quality control. President W. T. Chaurte, a graduate of Massachusetts Institute of Technology, undertook the task of developing a scientific basis for interpreting the results of tests on his product. This was the beginning of what is now recognized in this country as statistical control. In 1924 W. Hellmich, managing director of Normauschuss and co-director of the V. D. I., wrote to the American Standards Association, inquiring whether anything along the same lines had been done in the United States.

In the United States, the leading authority on the subject is Dr. W. A. Shewhart, of the Bell Telephone Laboratories. In addition to its introduction at the Bell Laboratories, this method of statistical control has been applied in the United States in the paint and varnish industry, the ceramic industry, and in the brass and alloys industry. R. F. Passano has done considerable work with it at the American Rolling Mill Co., and M. F. Skinker, assistant director of research, has introduced it at the Brooklyn Edison Co. In Canada, considerable work has been done at the Ontario Research Foundation.

As the subject is based upon the higher mathematics of the laws of probability and chance, the immediate task of those sponsoring the application of this scientific theory is to put the material in a form in which it can be understood readily and applied by the average engineer. Considerable progress has been made in this direction, and judging by the round-table discussion of the subject at the A.S.T.M. meeting, the forthcoming year should show greater progress.

A.S.M.E. Applied Mechanics National Meeting

Designing engineers will find much of value and interest in many of the papers presented at the Applied Mechanics National Meeting of the A.S.M.E., held at Yale University, New Haven, Conn., June, 23-25.

On the general subject of vibration, Professor F. M. Lewis, of the Webb Institute of Naval Architects, presented his paper on the subject "Vibration During Acceleration Through a Critical Speed," which discusses the subject of maximum amplitudes of vibration that might occur when an engine is passing through the critical speed. At the same session, J. G. Baker, Westinghouse Electric & Mfg. Co., spoke on the subject of "Self-Induced Vibrations," a practically new subject which attempts to explain some of the troubles in modern machinery and equipment.

Stresses in semi-circular plates, such as used in steam turbine diaphragms, were analyzed in a paper by Mr. A. M. Wahl, while the bending of long rectangular plates with lateral loading was the subject of a

paper by Professor S. Way of the University of Michigan. Dr. N. Goodier, Ontario Research Foundation, presented an analysis of the stress conditions around spherical holes or inclusions in members subjected to tension or twist. This ties up intimately with the effect of inclusions or holes in causing fatigue cracks to begin.

Papers dealing with elasticity and plasticity were presented by Dr. A. Nadai, H. Hencky, G. MacCullough, covering the subjects of "Creep of Metals," "Elastic Behaviour of Vulcanized Rubber," and "Creep in Bending," respectively. Professor J. H. A. Brahtz gave some interesting results of the investigation of stresses in the re-entrant corners of angles and I-beams. Professor M. A. Sadowsky presented a paper on "The Distribution of Pressure Concentrated on a Small Surface," and W. O. Richmond, of the Westinghouse Electric & Mfg. Co., gave a paper on the "Distribution of Stress Due to a Single Force Acting on the Edge of a Circular Hole." These last two papers were presented by titles.

At the Hydro- and Aerodynamics Session, O. Tietjens showed an interesting film on a new kind of high-speed water craft using fins to decrease water resistance and thereby attaining higher speed.

New Products Conference

Sometime during the latter part of September, the Cleveland Engineering Society will hold a "New Products Conference." The exact date, at present not set, together with the program for the meeting, will be announced in the September number.

American Standards Association Wire and Cable Standards

American Standards Association has approved three standards relating to wires and cables. These are: (1) American Standard Definitions and General Standards for Wires and Cables, (2) American Tentative Standard Specifications for Weather-proof (weather-resisting) Wires and Cables, (3) American Tentative Standard Specifications for Heat-Resisting Wires and Cables.

The first of these comprises a rearrangement of the A.I.E.E. standards No. 30 of October, 1928, which have been brought up to date with certain additions and the deletion of some of the tabular matter.

The specifications for Weather Proof Wires and Cables cover weather-proof wires and cables and the materials used for coverings and saturating compounds as applied to metallic conductors.

The specifications for Heat-Resisting Wires and Cables cover the usual types of heat-resisting coverings commonly known as "slow-burning" as applied to metallic conductors for use in hot dry locations where the other types of insulation would not endure or where the presence of large masses of inflammable materia's would be objectionable.

International Electrical Congress

Held at Paris from July 4 to 12, the International Electrical Congress attracted leading engineers from many nations. This meeting, to some extent, was to commemorate a similar one held in 1881, and was prompted by the feeling that there is a definite need for an international exchange of knowledge, experience and ideas.

In a total of about 200 papers for this meeting, all phases of electrical engineering, including the commercial aspects of electrical developments, were covered. Of course, time did not permit the presentation of all of these papers, and they are to be published in bound volumes so as to be available to everyone interested.

New Specification A48-32T for Cast Gray Iron

As the result of several years research work by the Gray Iron Institute, Cleveland, O., the old specification for gray iron castings known as A48-29T has been revised by the American Society for Testing Materials to a tentative form A48-32T.

Three new features of interest to designers are: (1) Classification of gray iron into 7 tensile groups: 20,000, 25,000, 30,000, 35,000, 40,000, 50,000 and 60,000 pounds per sq.in.; (2) Tensile and transverse bar sizes for standard testing increased in number from one to three (0.875 in., 1.2 in. and 2.0 in. diameter) in order to provide adequate design information pertaining to various sections in castings; (3) Chemical composition subordinated to physical properties.

It will now be possible for foundrymen to state specifically the strength which can be expected in critical sections of their castings. Heretofore the single arbitration test bar has not given results which could be applied satisfactorily throughout.

Meetings

A.S.M.E.

National Wood Industries meeting of the American Society of Mechanical Engineers is scheduled for Sept. 19-23, at Jamestown, N. Y., together with a commercial and educational exhibition. Calvin W. Rice, secretary, 29 West 39th St., New York City.

National Machine Shop Practice Meeting of the American Society of Mechanical Engineers will be held at Buffalo, N. Y., during the week of Oct. 3. Calvin W. Rice, secretary, 29 West 39th St., New York City.

A.S.S.T.

Sponsored by the American Society for Steel Treating, The National Metal Congress and Exposition will be held Oct. 3-7, 174th Regiment Armory, Buffalo, N. Y., with the American Society of Mechanical Engineers, Institute of Metals, American Institute of Mining Engineers, American Welding Society, and The Wire Association cooperating. W. H. Eisenmann, director, 7016 Euclid Ave., Cleveland.

S.A.E.

The Aeronautic Meeting of the Society of Automotive Engineers is scheduled for Aug. 30-31, at the Hotel Statler, Cleveland. John A. C. Warner, secretary, 29 West 39th St., New York City. Papers of particular interest to designers are:

"Vibration of Instrument Boards and Airplane Structures"—Stephen J. Zand;

"Late Developments in Airplane Stress Analysis Methods and Their Effect on Airplane Structures"—Richard C. Gazley;

"Hollow-Steel Propeller Blades"—J. H. McKee;

"Indicators as a Means of Improving Aircraft Engine Performance"—Ford L. Prescott;

"Transport Plane Design and Testing"—Frank Courtney;

"The Economic Aspects of Transport Airplane Design"—Ralph Damon.

Engineers and their Activities

A. A. Potter

ALFRED E. BALLIN has been elected vice president in charge of engineering of the Nordberg Mfg. Co., Milwaukee, Wis.

EUGENE L. BEECHER, who has been research engineer of the Gabriel Co., 1407 E. 40th St., Cleveland, Ohio, has been promoted to the post of chief engineer of that company.

R. GUY COWAN has been placed in charge of the new design-counsel service of Ferro Enamel Corp., Cleveland, O.

R. C. DARNELL has been appointed chief engineer of the Handy Governor Corp., 3925 W. Fort, Detroit.

JOHN A. HUNTER, professor of mechanical engineering, University of Colorado, has been nominated as one of the managers of the A.S.M.E. for 1933.

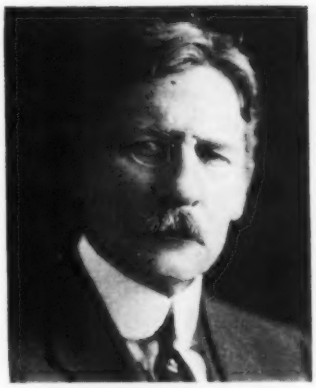
J. G. LANNING has been appointed chief engineer of the Hawley-Jones Corp., Corning, N. Y. He was formerly connected with the Detroit Lubricating Co. and with Manning, Maxwell & Moore.

GEORGE A. PAGE, JR., until recently design engineer of the Curtiss Aeroplane & Motor Co., Buffalo, N. Y., has been appointed chief engineer of the Curtiss-Wright Airplane Mfg. Co., Robertson, Mo.

DEAN A. A. POTTER, Schools of Engineering, Purdue University, has been nominated as president of the A.S.M.E. for 1933.

DR. H. M. RAYMOND, formerly dean of engineering, has resigned as president of Armour Institute of Technology, Chicago, Ill.

P. B. Rogers



Frank J. Sprague

Clifford A. M. Weber

P. B. ROGERS has been elected vice-president in charge of engineering and a member of the board of directors of Great Lakes Aircraft Corp., Cleveland, O.

ROBERT L. SACKETT, dean of engineering, Pennsylvania State College, has been nominated as one of the managers of the A.S.M.E. for 1933.

R. H. SMITH, associated with the Boston office of the Reliance Electric & Engineering Co., Cleveland, has been appointed manager of the company's development engineering department at the home office.

FRANK J. SPRAGUE, engineer, inventor and organizer of the Sprague Electric Elevator Co. and Sprague Electric Co. was honored by a special meeting of the A.S.M.E. at the Engineering Societies Auditorium, New York, on his 75th birthday, July 25. A distinguished committee had charge of the arrangements, and addresses were given by Dr. John H. Finley, Frank Hedley and Rear-Admiral S. S. Robison.

PERRY L. TENNEY, until recently chief engineer of the Muncie Products Division of General Motors Corp., has become engineer in the Transmission Division of the Olds Motor Works, Lansing, Mich.

HERBERT C. WALTERS, formerly chief engineer of the Delco Aviation Corp., has become a member of the engineering department of the Scintilla Magneto Co., Inc., Sidney, N. Y.

CLIFFORD A. M. WEBER, manager since 1931 of the Motor Appliance Engineering Department, East Springfield Works, Westinghouse Electric and Manufacturing Company, has been elected president of the Engineering Society of Western Massachusetts.

KARL H. WHITE, former design engineer, Curtiss-Wright Airplane Co., Robertson, Mo., has been appointed engineer in charge of the physical testing laboratory, Naval Aircraft Factory, Philadelphia, Pa.

JOHN G. WOOD, until recently general manager of the Muncie Products division of the General Motors Corp., has been appointed chief engineer, the Olds Motor Works, Lansing, Mich.

NEW MATERIALS AND PARTS

Philadelphia "Motorized" Vertical Speed Reducer

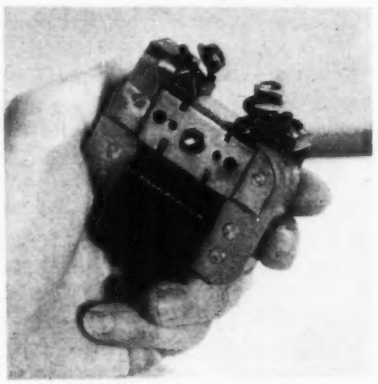
Philadelphia Gear Works, Philadelphia, Pa., announces a new "motorized" vertical drive speed reducer. Being self-contained, it requires no baseplate, flexible coupling or extra framework for mounting. The gears are of hardened nickel steel, cut with helical teeth; anti-friction bearings are used throughout. The slow-speed shaft is rigidly mounted to prevent whipping, and all moving parts operate in an oil bath. Special oil seals and stuffing box are used to overcome the possibility of oil leakage; the unit is noiseless in operation, without the use of non-metallic gears. The reducer is available in ratios from 3 to 1 up to 200 to 1, or higher if desired, and for horsepowers from ½ to 75.

Wagner Capacitor Motors

Supplementing its other lines of single-phase motors, the Wagner Electric Corp., 6444 Plymouth Ave., St. Louis, Mo., now offers a line of capacitor motors known as Type RZH. These are condenser-start, condenser-run, split-phase motors, the condenser capacity being varied from start to run by means of a switch and auto-transformer. In appearance these motors differ from others in the line in that a box is mounted on top of the motor to accommodate the condenser and auto-transformer. Capacitor motors are available as rigid-mounted or rubber-mounted types.

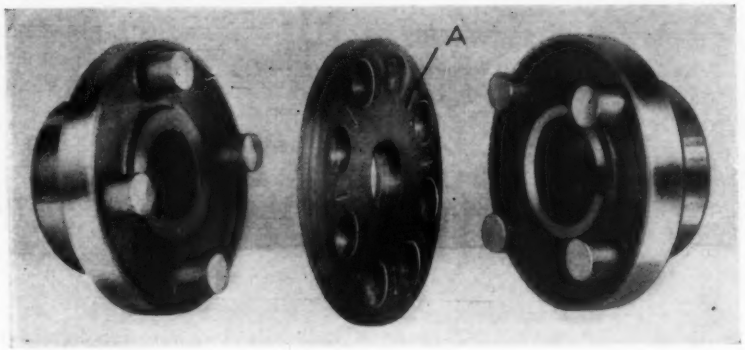
Barcol Reversible Midget Shaded Pole Motor

A reversing model of the Barcol midget motor, described in the June *Product Engineering,* is now made available by Barber-Colman Co., Rockford, Ill. Unidirectional models of this fractional-horsepower, single-phase induction motor of the shaded pole type make use of shading rings to introduce a lag in part of the field surrounding the rotor and thus to obtain a rotating field which produces motor action. The new reversing model employs shading coils in place of the shading rings of the other models; the motor is connected with current on the main field coil all the time, no rotation being produced until current is sent through one or the other of the two sets of shading coils. Direction of rotation is determined by which of the two sets of shading coils is energized. These coils are made the same as the main field coil of high grade enameled wire on a phenolic resin spool and impregnated with moisture-proof varnish. The leads are brought out to plates mounted above the coils; control leads as required are taken off from these plates. In all other respects the reversing model is identical with the unidirectional type. Available in several sizes to suit various applications, this model is claimed to have high starting torque, high power considering the size of the motor, and relatively high efficiency.



Chapmanizing

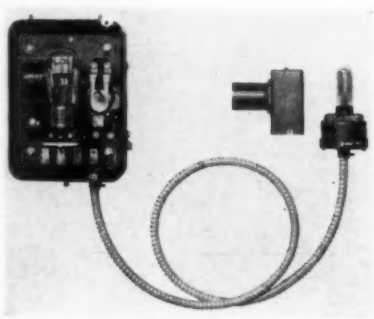
Under the name of "Chapmanizing," a new method of casehardening ordinary carbon steel has been developed by Chapman Valve Mfg. Co., Indian Orchard, Mass. In principle it is claimed that all forged or cast-steel parts, from very small objects to individual pieces as large as 24 in. diameter, 10 in. deep, can be treated to a Brinell hardness ranging from 700 to 900. With certain types of steel and a slight variation in the process, a Brinell of 900 to 1100 can be produced. The usual depth of the



case is 0.025 in. The specifications of steel particularly recommended for the application of this treatment are SAE 1020, 1025, 1112, 1120, 1315, 3115, 3220, 6115, 6120. It is claimed that the growth under treatment will not in any case exceed 0.001 in. and that the maximum temperature rating of the steel will not be changed. Although it is not necessary to have machined surfaces in order to Chapmanize, wear resistance and hardness will be much greater in surfaces that have been machined and ground. Machining before treatment is therefore recommended in all cases where excessive wear is to take place.

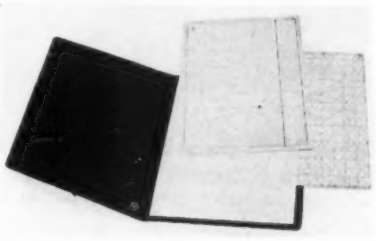
General Electric Photoelectric Relay

A general purpose photoelectric relay with several new features has been developed by General Electric Co. Provided with a more sensitive control than the models which it supersedes, this relay will permit operation at a minimum light intensity of three foot-candles. Ten foot-candles was the minimum light intensity prescribed for the old model. Other features offered by the new relay include improved dielectric strength through the use of a Bakelite panel, an improved phototube housing with a cast aluminum base and a drawn aluminum hood, reduced capacity leakage due to the use of flexible armored cable between the phototube and the relay proper, thus affording better operation in damp atmospheres. The relay uses a Pliotron tube specially designed for industrial purposes, such as operation of magnetic counters, control of mechanisms, starting and stopping machinery.



E-Z-Draft

McCauley & Madison, Inc., Detroit, Mich., has copyrighted and is marketing a useful designer's sketch pad which combines a durable artificial leather cover, a 100-sheet pad of special transparent paper and ruled bristol board plates to be inserted under the top sheet of paper. One plate is ruled isometrically on one side, and in standard $\frac{1}{8}$-in. cross section on the other, either side having horizontal and vertical lines at 1 in. intervals. On the isometric side there are two other series of parallel lines running to the right and left, these lines being sloped at an angle of 30-deg. which constitutes the isometric part of the plate. The perspective plate is designed in the same size with lines running vertically which diminish in space to the right and left, and is designed with a series of lines which converge at the right and left, thereby determining specific vanishing points. Since the paper in the pad bears no cross lines whatsoever, excellent blueprints can be made therefrom. The E-Z-Draft is obtainable in four sizes ranging from 9x12 in. to 24x36 in.



Brown Buffer-Slot Coupling

Buffer-slots cut at regular intervals in the flexible, laminated leather disk of the "Buffer-Slot" coupling give flexibility and permit the coupling to be used under conditions of angular and parallel misalignment and shock loads. This coupling, developed by the Brown Engineering Co., Reading, Pa., can be run in either direction. Driving lugs extend through the buffer-slot disk *A* and are claimed not to wear ridges that would cause end-play when the coupling is operated for a long time in one direction. The leather between the driving lugs transmits power through compression and not through tension. It also electrically insulates the two connected units from one another. Under a sudden overload, all the buffer-slots in the leather disk close automatically, and when the load becomes normal they reopen.

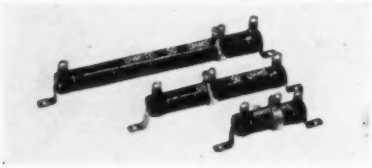
Granodine

The Granode process announced by American Chemical Paint Co., Ambler, Pa., is a rustproofing process which produces a thin, gray-colored undercoat for varnishes and lacquers. It is not a claiming preparation; the metal must be precleaned thoroughly to remove oil and rust. Granodine itself is an active, heavily pigmented paste which is diluted with water to spraying consistency. The resulting coating is said to be smooth, practically insoluble in water, tough enough to withstand mild abrasive action, and practically inseparable from the metal. The manufacturer states that it will very shortly be in a posion to ship Granodine in powdered form as well as in paste.

Ohmite "Dividohm" Resistors

Ohmite Mfg. Co., 636 N. Albany Ave., Chicago, Ill., announces that its line of semi-variable resistors has been increased to include values up to 100,000 ohms. A new feature known as the percentage-of-resistance scale has

been adopted as standard on all the units, of this type. The scale, which is marked on the side of the unit, permits setting adjustable lugs to any desired resistance without complicated meter readings. The resistors



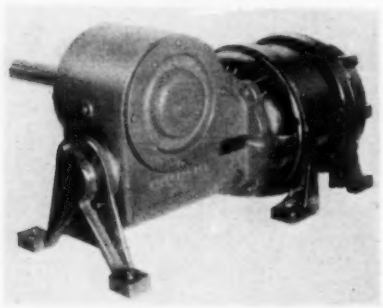
are of the vitreous enameled, porcelain tube type. The resistance wire is wound around an Isolantite tube and covered with a vitreous enamel which is given a Class C rating by the Underwriters' Laboratory. A narrow strip along one side of the unit is left exposed so that contact may be made at any point by means of adjustable lugs. As over four-fifths of the surface is covered with enamel, the unit is protected from both mechanical and electrical injury.

"Dividohm" units are made in three sizes, 2 in., 4 in., and 6 in., which are rated at 30 watts, 55 watts, and 75 watts respectively. The resistance values range from 1 ohm to 100,000 ohms. It is said that these units may be used safely at their full rated values when installed where there is normal air circulation.

Millholland Motorized Speed Reducers

The motorized speed reducers developed by the Millholland Corporation, 1145 E. 22d St., Indanapolis, Ind., use standard motors without special bearings, shafts or end bells. This is accomplished by bolting the reducer case directly onto the machined register of a standard motor end bell.

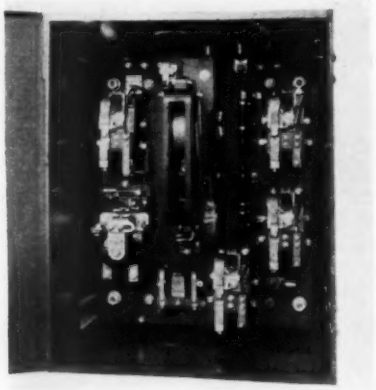
The speed reducers are of the worm and worm gear type, with anti-friction bearings supporting both.



The design provides for an outboard foot support to the reducer. Hollow levelling thimbles, through which the reducer hold-down screws are placed, are provided in the outboard-support feet to accommodate variations in the center height of the motors. The output shaft is made to project in any one of four positions in two planes, and the position may be changed without the use of special tools. Air vents in the reducer case at the motor end allow the motor to receive cooling air. Sizes are available to fit all standard motor frames from $\frac{1}{4}$ to 10 hp.

Westinghouse Non-Reversing Time Limit Starter

Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., has recently announced a new type starter for heavy duty with non-reversing, constant-speed, d.c. motors on pumps, fans, hoists, all kinds of conveyors, loading machines and all types of



mining machines. This starter gives definite time limit acceleration without the necessity of intermediate relays. It is said that it gives a compact, simple and definite time limit method for accelerating a motor over the greatest voltage range for which a magnetic starter has been built.

The "Timetactors," one of which is used for each step of resistance, will operate and cut out the accelerating resistor on any voltage which will hold the line contactor closed, because the Timetactor is electrically opened and mechanically closed. New heavy-duty contactors of high arc-rupturing capacity provide short circuit protection. Thermal and magnetic overload protection is provided. Either hand or automatic reset is available. It is claimed that the starter will operate perfectly under conditions where voltage before starting the motor is approximately 65 per cent of rated voltage and then drops to 25 per cent as soon as the motor starts. The 75 hp., 230 volt, four-step starter in the illustration is mounted in a cabinet $19\frac{1}{8}$ in. wide, $24\frac{7}{8}$ in. high by $22\frac{1}{8}$ in. deep.

Ingoclad Two-Ply Stainless Sheet Steel

Advance information on a two-ply stainless sheet steel produced by a patented process from the composite ingot is released by Ingersoll Steel & Disc Co. (division of the Borg-Warner Corp.), Chicago, Ill. It is claimed that the manufacturing process has been developed to a point where a perfect bond between the stainless steel and the carbon steel back is assured. The new material may be deep-drawn, stamped, welded, formed and polished. It is now being produced in various gages and sizes and will be marketed at a price which will permit its use for applications which have heretofore been forced to forego solid stainless steels.

WHS Two-Speed Reducer

To meet the needs of conveyor and other manufacturers requiring a change of speed under load in their drives, Winfield H. Smith, Inc., Springville, Erie County, N. Y., has designed a two-speed reducer obtainable in ratios of reduction ranging from 4:1 to 150:1. A neutral position is provided whereby the mechanism operated by the slow-speed shaft can be stopped without shutting off the power to the unit. On the high-speed shaft is mounted a hardened steel worm driving a special alloy bronze worm gear on the inter-

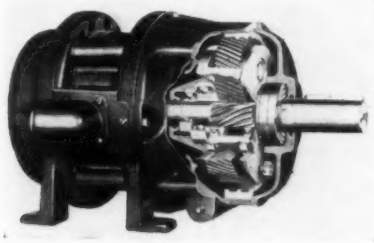
mediate shaft. On this intermediate are also mounted two spur gears which are in constant mesh with two spur gears of special alloy metal mounted on the slow-speed shaft. The drive is taken by means of a four-jaw clutch engaging with one or the other of the spur gears on the slow-speed shaft. This clutch is actuated by a yoke arm extending through the top of the reducer to which a shifting arm is attached. It is claimed that the unit can be shifted easily from slow-speed to high-speed operation or to a neutral position while operating under a full load.

Century Electric Capacitor Motors

Supplementing its other lines of single-phase motors, Century Electric Co., St. Louis, Mo., has offered a line of capacitor motors in sizes from $\frac{1}{16}$ to 10 hp., furnished for single or multispeed operation with normal, high or low torque characteristics, to meet requirements of each particular application. Small sizes have the capacitor box mounted above the motor, although the boxes can be furnished for wall mounting as in larger sizes. Integral sizes have a separate capacitor box, which may be furnished for either wall or floor mounting. Operating characteristics cover a wide range of applications from direct-connected fan to more exacting services.

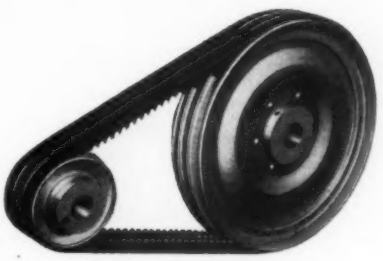
General Electric Gear-Motors

General Electric Co., Schenectady, N. Y., announces a new line of "gear-motors" readily adaptable for application to machinery of widely varying designs. These consist of normal speed motors in combination with built-in, internal-helical, planetary-gear speed-reducers. Since this type of construction permits a wide choice of speeds on the output shaft, it is possible to adapt a gear-motor to almost every type of low-speed drive with high efficiency. Gear-motors are also available with special electrical characteristics, such as high starting torque with low starting current, normal starting torque with low starting current, adjustable, varying speed, multispeed, etc., or mechanical features such as totally inclosed; totally inclosed, fan-cooled; or Class I, Group D, construction for use in hazardous gas locations. A.c., polyphase, squirrel-cage and wound-rotor gear-motors are available in ratings up to 75 hp., single phase to 5 hp., and d.c. to $7\frac{1}{2}$ hp. D.c. gear-motors are offered for those applications where either the advantages of adjustable speed drive or a d.c. source of supply render their use desirable. Standard gear-motors can be had with shaft speeds down to 13 r.p.m.



"Day-Steel" V-Groove Pulleys

A new line of "Day-Steel" single and multiple V-groove pulleys for use with Dayton cog-belts is announced by Dayton Rubber Mfg. Co., Dayton, Ohio. These pulleys, designed for ratings of $7\frac{1}{2}$ hp. and below, are accurately formed from heavy gage pressed steel for maximum strength, welded both at the rim and at the web, and rigidly assembled with a hub of unique construction to give an accurately balanced and true running pulley.



Reliance A.C. Motors for Rapid Reversals

Alternating-current motors recently developed by the Reliance Electric & Engineering Co., 1042 Ivanhoe Road, Cleveland, Ohio, may be applied to machine tools such as tapping machines, broaches and motor-operated chucks. Being designed with torque characteristics to give rapid reversals, these motors may be built to reverse 40 times per minute. Low-inertia rotors are provided to keep reversing losses at a minimum. The motors are fully inclosed; internal and external fans are both made of cast duralumin.

"Stearns" Small Magnetic Clutch

The Magnetic Manufacturing Co., Milwaukee, Wis., has developed a very small size mu'tiple-disk mag-

netic clutch. This unit measures only $4\frac{1}{2}$ in. in diameter and requires 30 watts d.c. for operation. The torque capacity is 600 in.-lb. The magnet is stationary, the clutch being designed for use as a built-in machine tool drive.

Copperface and Zincface Strip Steel

Marketed by Joseph T. Ryerson & Son, Inc., Chicago, Ill., for Pittsburgh Cold Rolled Steel Co., "Copperface" is a cold rolled strip steel coated with pure copper. The strip is passed through a special coating machine and covered with a copper mix made up of pure copper powder and fluxes carried in a liquid medium. The strip passes continuously through a furnace at a temperature of approximately 2,000 deg. F. By this process the copper penetrates the pores of the steel, forming an alloy which cannot be removed except by abrasion.

Since in a great many ways Copperface has the physical characteristics of pure copper, it can be used to replace a certain amount of copper and brass. The surface acts as a lubricant so that in drawing or forming some manufacturers have been able to eliminate extra drawing operations or

intermediate annealings. The material can be spot-welded without injuring the surface. It tins and solders as readily as pure copper and is a good base for nickel and chrome and other platings. It has enough copper on the surface so that it can be hydrogen brazed successfully.

"Zincface," like Copperface, is cold rolled strip steel, coated with pure zinc by substantially the same process but at a lower temperature. It is claimed that a very rust resistant zinc-iron alloy results which will stand severe forming without separating the zinc from the steel. The pure zinc surface furnishes a good foundation for paint, enamel and lacquers.

New Books and Publications

"BERYLLIUM—ITS PRODUCTION AND APPLICATION"

[*Siemens-Konzern. Translated from the German by Richard Rimbach and A. J. Michel.*] 331 pages, 6x9 in. Clothboard covers. Indexed and illustrated. Published by The Chemical Catalog Co., 419 Fourth Ave., New York, N. Y. Price $10.

Incorporating as it does the work of the Central Committee on Scientific and Engineering Research of the Siemens Company up to 1929, this excellent book is a compendium of German experience with beryllium and its alloys. Twenty-two engineers, chemists, physicists, metallurgists and other scientists collaborated in its preparation. Thus between one pair of covers the interested engineer will find the subject of this revolutionary element thoroughly covered from the German point of view. He should, however, be warned that certain recent American developments promise to modify and even to upset some of the conclusions therein contained.

"RAMBLING THROUGH SCIENCE"

[*A. L. DeLeeuw, M.S., M.E.*] 320 pages, 5½x8 inches. Illustrated by diagrams. Bound in cloth boards. Published by Whittlesey House, McGraw-Hill Book Co., Inc., New York and London. Price $2.50.

Although Mr. DeLeeuw's book is intended for popular consumption it should be welcome to engineers and other industrial executives who are interested in the advances of science because of their ultimate, and often immediate effect on industry. The author, by education a physicist and by training and experience an engineer, brings to his task of interpreting science an unusual combination of theory, practice and a conversational literary style. He makes the fourth dimension seem obvious and he "rambles" widely from atoms to astral mechanics, and from light waves to the quantum theory. The reader cannot help acquiring a new and higher opinion of science and scientists as he "rambles" with Mr. DeLeeuw.

Ball Bearings

[*Auburn Ball Bearing Co., 57-85 Clarissa St., Rochester, N. Y.*]. This 36-page reference manual describes the bearings, gives mounting instructions, dimension limits, lists of stock sizes and capacity tables. Special bearings are also illustrated.

Cadmium Plating

[*Udylite Process Co., Detroit, Mich.*] "More Than a Pound" classifies the numerous engineering and laboratory services available to users of the Udylite cadmium plating process.

Circuit Breakers

[*Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.*] Circular 1939 contains a general description of "De-ion" circuit breakers, breaker ratings, and their applications. Circular 1937 gives construction and application of the 225-amp. breaker.

Catalog 2246 presents full details of the operation, construction and application of each of the many Nofuze "De-ion" circuit breakers, and includes prices for the various ratings and suggested specifications.

Electrical Controls

[*Detroit Lubricator Co., Detroit, Mich.*] Catalog G describes "Mercoid" controls of the following types: level-operated; temperature, differential; liquid level; pressure; vacuum.

[*Providence Electric Controls, Inc., Providence, R. I.*] "Mercury Contact Switches" gives specifications of the "Magnetrole" mercury switch in sizes from 2 amp., 125 volts, to 25 amp., 50 volts.

Electrical Measuring Instruments

[*Leeds & Northrup Company, Philadelphia, Pa.*] "Temperature Measurements in Generators, Transformers and Cable Systems" is a revised edition of Bulletin No. 871. Included is a list of all the bulletins issued by the company.

Induction Motors

[*Reliance Electric & Engineering Co., Cleveland, O.*] Bulletin 111 illustrates and describes the manufacture and general characteristics of Type AA Reliance squirrel-cage induction motor, ½ to 200 hp. Bulletin 112 does the same for Type AA Form F fully-inclosed fan-cooled induction motors.

Pumps

[*Geo. D. Roper Corp., Rockford, Ill.*] A new folder shows all types of Roper hand- and power-operated rotary pumps.

Rolled and Formed Sections

[*Truscon Steel Co., Youngstown, O.*] Handbook N-20 lists rolled and formed sections available in steel, aluminum, duralumin, monel metal, pure irons, special nickels, Everdur, bronze and other commercial metals.

Screw Threads

[*Dardelet Threadlock Corp., New York, N. Y.*] Bulletin No. 5 appears in its third edition as "Dardelet, the Self-Locking Screw Thread." Previous data on the thread and its applications have been augmented and revised.

Small Motors

[*Wagner Electric Corp., St. Louis, Mo.*] Bulletin No. 167, "Small Motors," is divided into eight parts describing single-phase, polyphase, and direct-current motors in fractional hp. ratings. Since it is in looseleaf form it may be kept up to date.

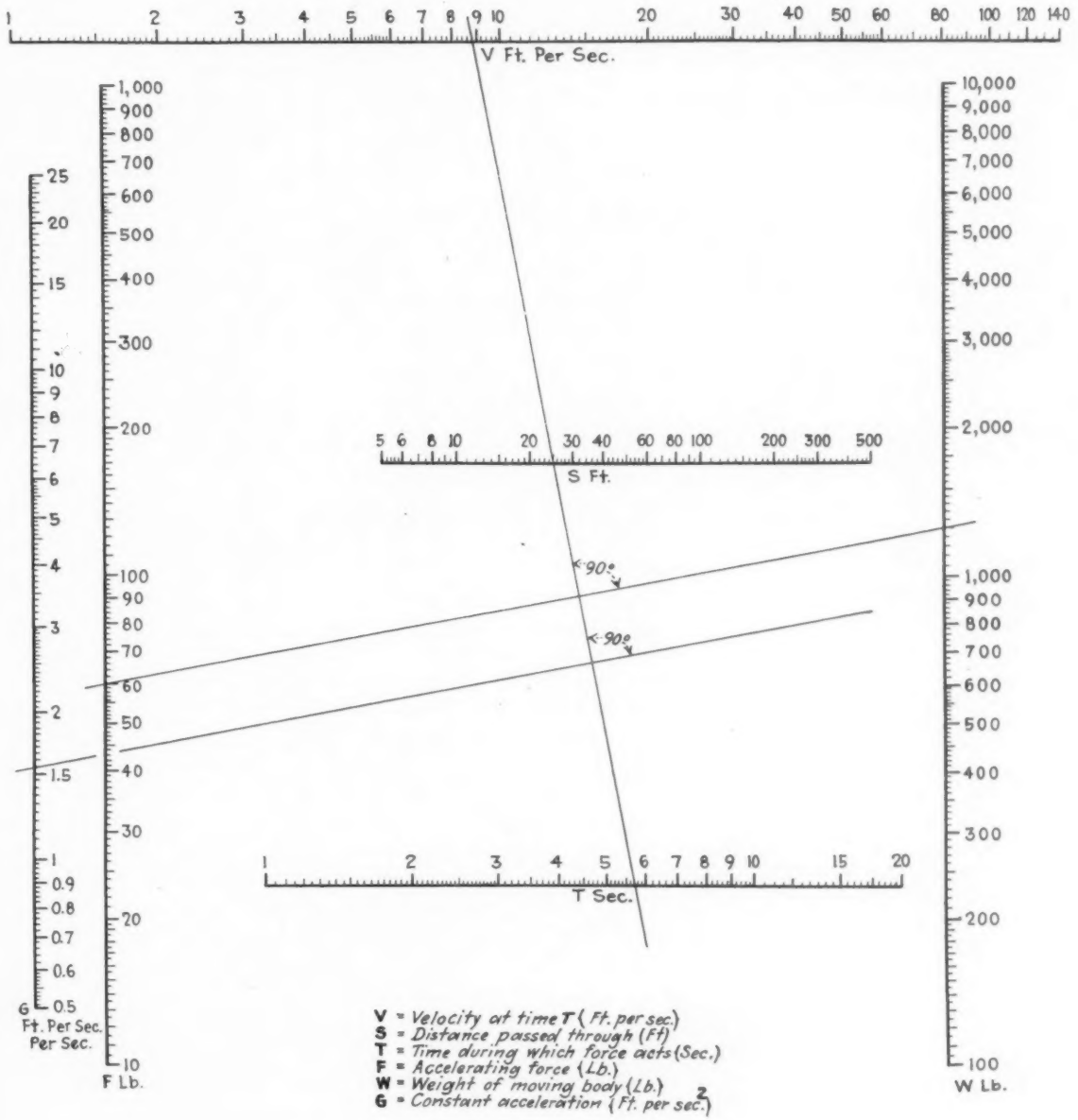
Stainless Steel Castings

[*The Cooper Alloy Foundry Co., Elizabeth, N. J.*] "Stainless Steel Castings" gives analyses, uses, examples, and a table comparing the physical characteristics of 18-8 stainless with brass, bronze, monel metal, carbon steel, and other metals.

Accelerated Linear Motion

Solution of the formulas:

$$\frac{2S}{T^2} = \frac{V}{2S} = \frac{V}{T} = \frac{32.16\,F}{W} = G$$



V = Velocity at time **T** (Ft. per sec.)
S = Distance passed through (Ft.)
T = Time during which force acts (Sec.)
F = Accelerating force (Lb.)
W = Weight of moving body (Lb.)
G = Constant acceleration (Ft. per sec.)^2

Contributed by M. G. Van Voorhis
Lakewood, Ohio

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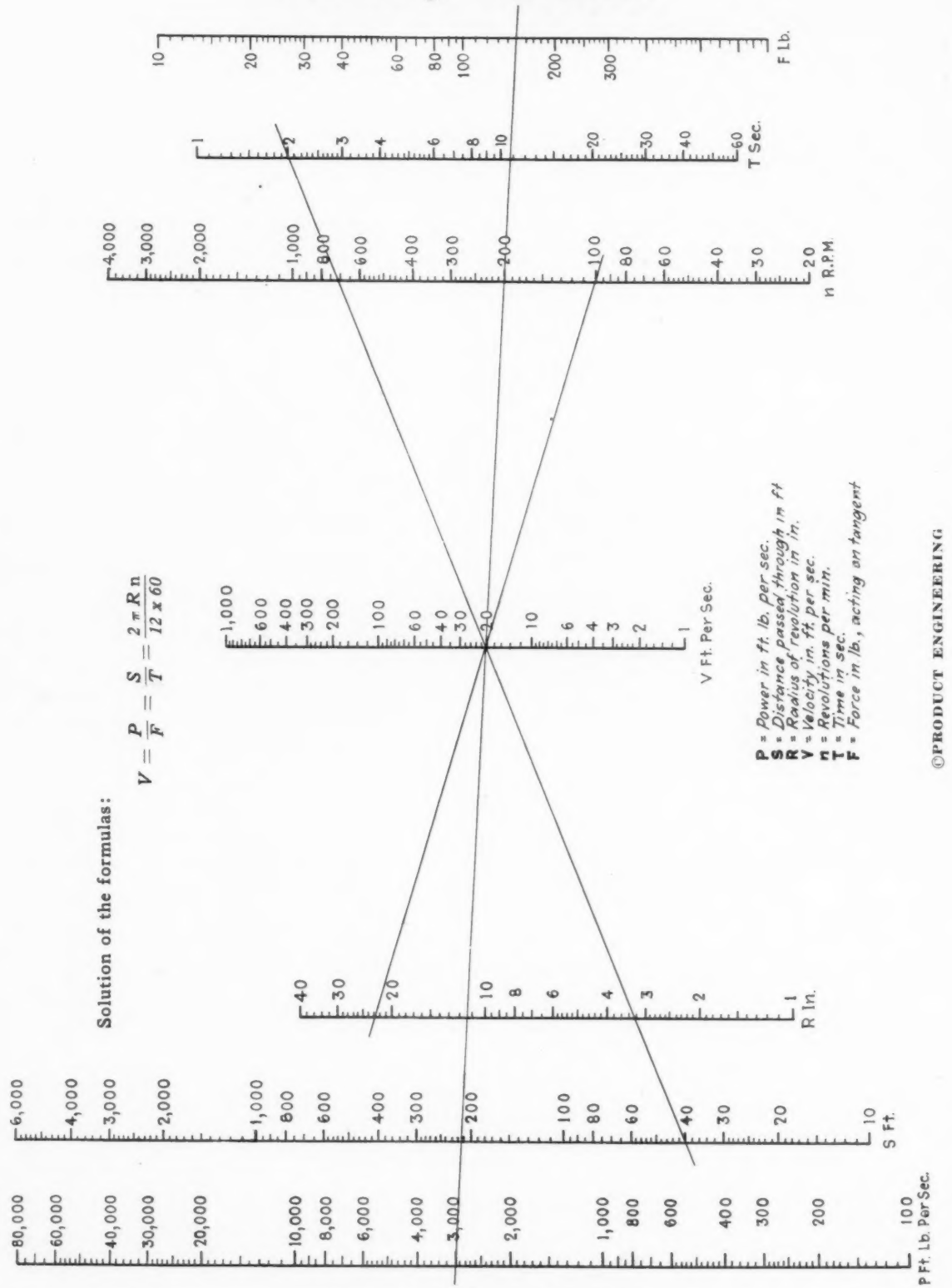
REFERENCE BOOK SHEET

Rotary Motion


F Lb.
T Sec.
n R.P.M
V Ft. Per Sec.
R In.
S Ft.
P Ft. Lb. Per Sec.
P = Power in ft. lb. per sec.
S = Distance passed through in ft
R = Radius of revolution in in.
V = Velocity in ft. per sec.
n = Revolutions per min.
T = Time in sec.
F = Force in lb., acting on tangent
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Solution of the formulas:

$$V = \frac{P}{F} = \frac{S}{T} = \frac{2\pi R\,n}{12 \times 60}$$

REFERENCE BOOK SHEET